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SECOND QUARTERLY REPORT

BRAYTON-CYCLE TURBOMACHINERY ROLLING-ELEMENT BEARING SYSTEM

prepared for

National Aeronautics and Space Administration

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EAST HARTFORD • CONNECTICUT

FOREWORD

This report describes the progress of work conducted between October 2, 1965 and January 2, 1966 by the Pratt & Whitney Aircraft Division of United Aircraft Corporation, East Hartford, Connecticut, on Contract NAS3-7635, Brayton-Cycle Turbomachinery Roller-Contact Bearings, for the Lewis Research Center of the National Aeronautics and Space Administration. The objective of the program is to design and demonstrate performance of a rolling-element bearing system for the Brayton-cycle turbomachinery being developed on Contracts NAS3-4179 and NAS3-6013.

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I. SUMMARY

This report presents the work completed during the second three-month period of a design and experimental program formulated to investigate the potential of oil-lubricated rolling-element bearings for Brayton-cycle space power machinery. The technical progress, the work accomplished and the program status for the second quarterly period from October 2, 1965 through January 2, 1966 are presented.

During the previous report period a rolling-element bearing lubrication system concept was evolved and during this period the concept was refined and defined further. Alternate concepts were also investigated but, while the alternates can provide greater assurance of satisfactory operation in all of the anticipated environments, each involves significantly high parasitic losses to the system. Features of the turbine-compressor rotor support and face seals were established, leaving only details to be worked out in other areas to complete the design phase of the turbine-compressor. The balancing requirements, material selections and related information for the turboalternator were specified, essentially completing the turboalternator design also. Laboratory scale tests of candidate adsorber materials indicate that a practical adsorber can be made for the conditions anticipated in the system. A five-ring polyphenyl ether was recommended to the NASA for the system lubricant.

II. INTRODUCTION

In the application of the Brayton cycle to space power sources, two types of rotor support systems are being considered and evaluated by the National Aeronautics and Space Administration. In one case, the rotating components are supported by gas bearings and the cycle working fluid is used as lubricant and bearing coolant. In the other case, the rotors are supported by rolling-element bearings which are lubricated and cooled by oil.

The specific Brayton-cycle machinery considered in this program consists of a turbine-driven compressor and a turbine-driven alternator. These units are being designed and constructed utilizing gas bearings under Contracts NAS3-4179 and NAS3-6013. The gas bearing version of this machinery is shown in Figure 1. The cycle flow enters the compressor at 76°F through the inlet duct around the housing which contains a thrust bearing and a radial bearing. The argon is compressed in the six-stage axial-flow compressor and then flows through the radial diffuser exit scroll and ducting. The argon is heated to 1490°F outside of the unit and is returned to the turbine inlet ducting and scroll. A second radial gas bearing is located between the compressor exit and turbine inlet. The argon flows through the single-stage axial-flow turbine which drives the compressor at 50,000 rpm. It then exhausts through the exit ducting which connects to the two-stage alternator-drive turbine. The 4-pole alternator is driven at 12,000 rpm and is supported by bearings on each side. After passing through the alternator-drive turbine, the argon exhausts through the exit scroll and ducting. It is cooled outside of the machinery and returned to the compressor inlet.

The vast majority of Brayton-cycle powerplants in use in the world today employ rolling-contact oil-lubricated bearings. In the application of Brayton-cycle machinery to the space environment, several significant questions require investigation to determine if rolling-element bearings can provide a satisfactory rotor support system for this application:

1) Can rolling-element bearings provide the endurance capability with the high reliability required in space applications for a nominal mission time of 10,000 hours?

2) Can the lubricant be cooled and circulated in a zero-gravity environment?

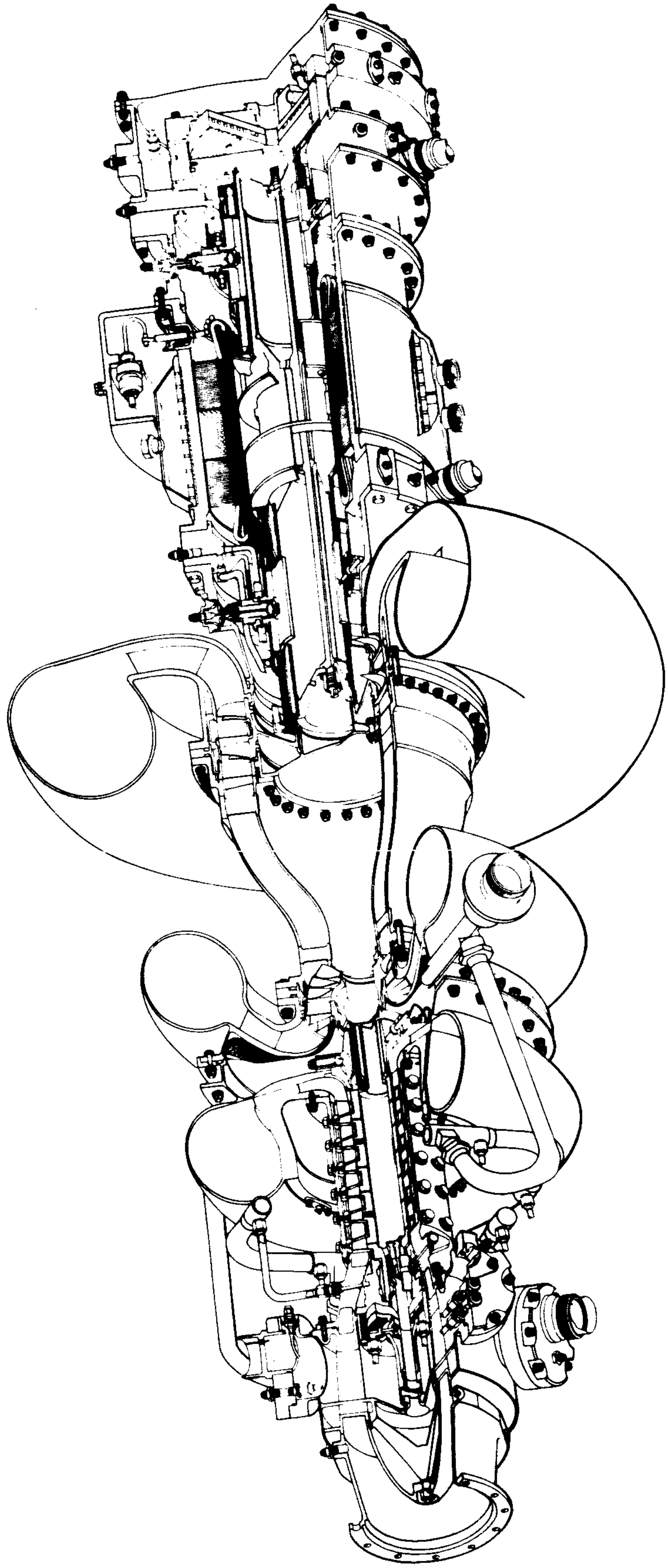


Figure 1 Brayton-Cycle Turbomachinery Employing Gas Bearings (Contracts NAS3-4179 and NAS3-6013)

3) Can the lubricant system and bearing cavities be sealed with low parasitic power losses by components with the required life capability?

4) Can the lubricant be prevented from contaminating the cycle working fluid (argon)?

The purpose of this program is to design and investigate the performance of a rolling-element bearing system for the turbine-compressor and turboalternator being developed under Contracts NAS3-4179 and NAS3-6013. The components of this rolling-element bearing system are bearings, seals, lubrication system and gas cleanup system. The work consists of the following four phases:

1) Design of a rolling-element bearing system that retains components of the turbine-compressor and turboalternator (designed under Contracts NAS3-4179 and NAS3-6013) to as great an extent as practical with no alteration in aerodynamic design. The shaft support system is intended to achieve low parasitic losses commensurate with high reliability for the full mission life.

2) Design and fabrication of component test rigs.

3) Conduct bearing, seal, scavenge separator and adsorber performance tests.

4) Conduct a pilot endurance demonstration of the rolling-element bearing system.

Although the rolling-element bearing system encompasses the turboalternator as well as the turbine-compressor, only the rolling-element bearing system components for the turbine-compressor will be investigated experimentally. The separator, which is an integral part of the turboalternator, will be evaluated in a separate test rig as will the residual oil adsorber.

III. ROLLING-ELEMENT BEARING LUBRICATION SYSTEM

The basic objective of the rolling-element bearing lubrication system for the Brayton-cycle turbomachinery is to provide proper lubrication and cooling while preventing contamination of the basic cycle working fluid. In addition, the bearing,seals, and lubrication system must not impose excessive losses on the powerplant. The lubrication system must be able to function properly in the gravity-free space environment and also on the ground for development testing and pre-launch checkout. During ground operation, the machinery may be oriented horizontally or vertically with the compressor end up.

The requirements that the powerplant be able to operate at various attitudes in various gravitational fields and also provide low losses presents a severe oil feed and scavenge system design problem. A conventional oil sump and oil pump system is dependent upon gravity and cannot be considered for this application. Therefore, systems which depend upon centrifugal effects to scavenge the oil and argon from the various bearing compartments have been considered. These same centrifugal slingers and pumps have been used to provide the pressure necessary to recirculate the oil through the system. A combination of seals is required to separate the lubrication system from the main cycle argon system. However, some argon will leak into the lubrication system and argon also must be circulated in the lubrication system, leading to the complication of two-phase flows.

In the previous report[1] a schematic diagram of a lubrication system was presented which used centrifugal principles to circulate the oil and argon in two-phase flow through the system. During the present report period this schematic was refined and alternate schematics were investigated. In all of the systems considered, certain basic features were retained. Each bearing is cooled by lubricant flowing under its inner race and lubricated by oil mist in argon gas passing through the bearing. Some main cycle gas is supplied in labyrinth seals upstream of face seals to provide proper pressure on the face seals and to prevent any oil that weeps past the seal from reaching the primary cycle. Provisions are incorporated to permit oil to lubricate the rubbing area of the face seal, if required. Oil is separated from the argon gas which leads to the main cycle in two stages. First, a centrifugal separator separates oil droplets and returns them to the lubrication system. The oil-argon mixture is cooled in the separator to condense some of the

[1] Numbered references are listed in Appendix 1

oil vapor. Then the relatively clean argon containing oil vapor passes through a molecular sieve where the oil is trapped and the resulting clean argon returns to the primary cycle. In the centrifugal separator, oil is returned to an accumulator by scooping the oil from a rotating pool. The heat generated in the bearings and seals is removed by a cooler located in the recirculating system.

The schematic diagram discussed in Reference 1 is presented in Figure 2. This system includes the features described above and utilizes centrifugal pumps in the seal plates of the turbine-compressor and turboalternator to furnish the necessary pressure to transport oil and argon through the system. The pressure rise in these pumps is low because the impeller velocities are limited by the available space. Therefore, centrifugal pumps in the turbine-compressor and turbo-alternator are arranged in series in order to provide the maximum system pressure rise. The oil flow to the turboalternator bearings is metered within the shaft and the same quantity of oil flows through the turbine-compressor bearing compartments. This system provides fairly low losses; however, it has several disadvantages. First, it limits the Brayton-cycle machinery to a simultaneous startup so that the turboalternator must be started at the same time as the turbine-compressor. Second, all of the oil passes through the turboalternator rotor which requires energy to increase the oil velocity to the impeller tip velocity. Actually, the turboalternator could be operated with about one-quarter of this oil flow. Third, a dead-ended accumulator is used and the possibility of gas accumulating in this container exists in the 10,000-hour machine. Fourth, when the system is operated in the vertical position in a 1 g field, approximately half of the oil must be carried vertically by about half of the argon flow through a vertical rise of 33 inches from the turboalternator to the turbine-compressor. This vertical rise results in increased pressure drop in the pipe and reduced argon flow attempting to carry the oil. As a result, slug flow would probably result in vertical operation. While slug flow might operate satisfactorily, it is unsteady and unpredictable. Therefore, slug flow is considered undesirable.

A modified system shown on Figure 3 was considered which incorporates a positive oil feed line from the accumulator to the turbine-compressor. In this system, the turbine-compressor can be started by injecting oil from the accumulator prior to starting the turboalternator. There is a positive oil flow through the accumulator which will minimize the possibility of trapping gas in the accumulator. The oil supplied to the turbo-alternator bearings is significantly reduced and only this flow is lifted during vertical operation. The combination of losses in the upward flow direction is improved due to the lower oil flow.

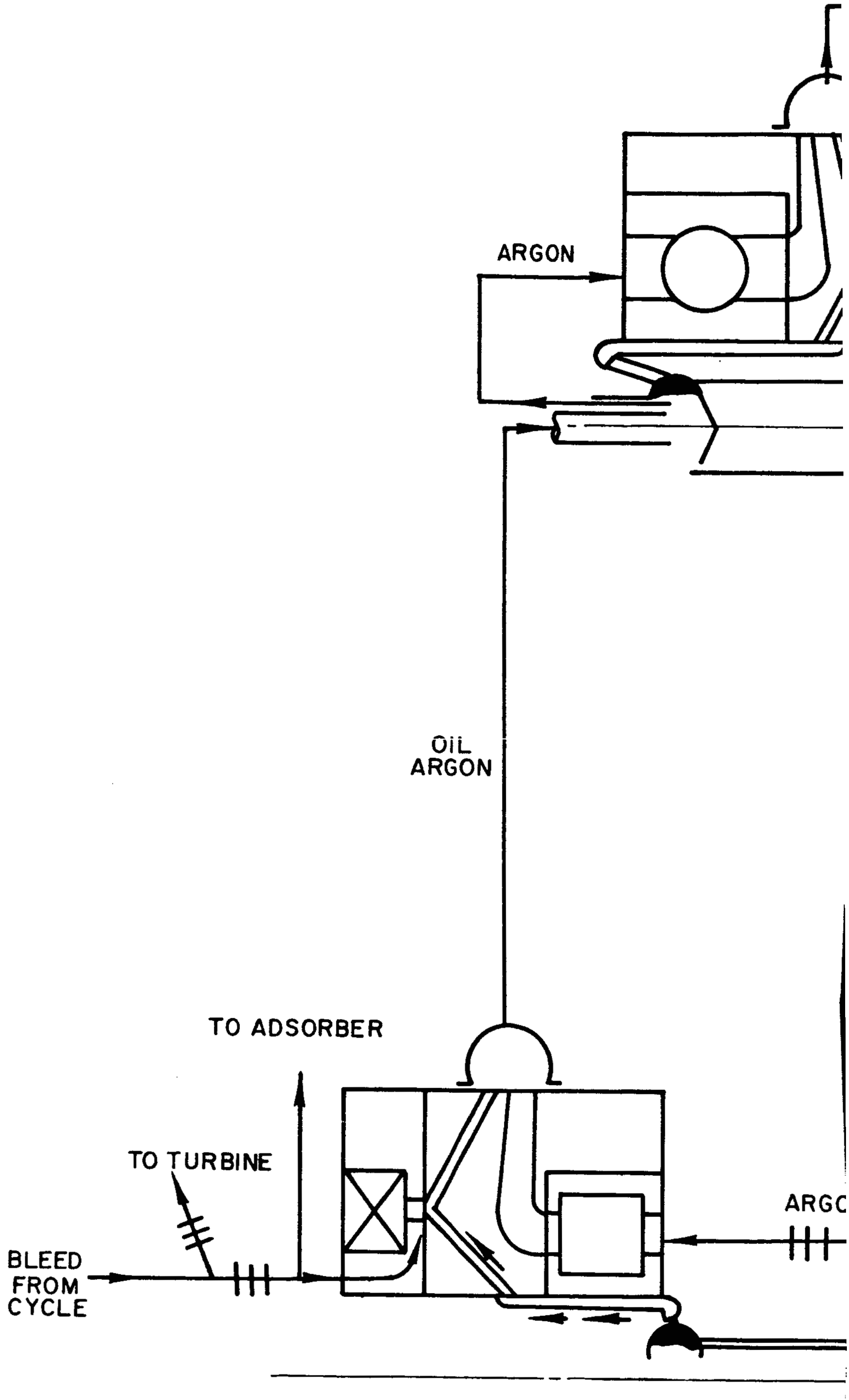
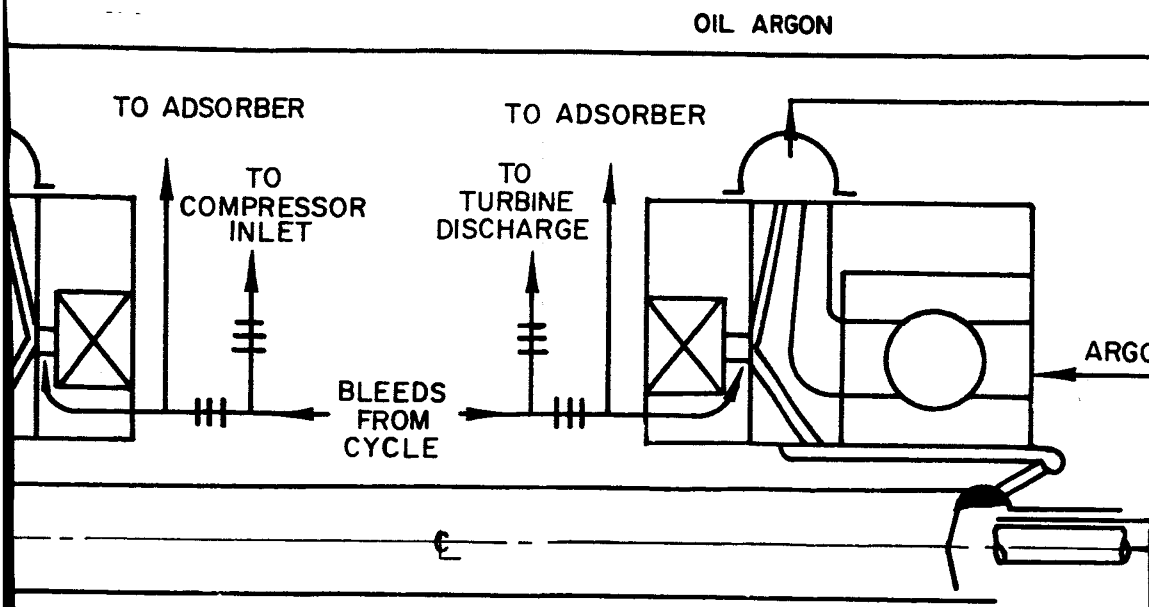
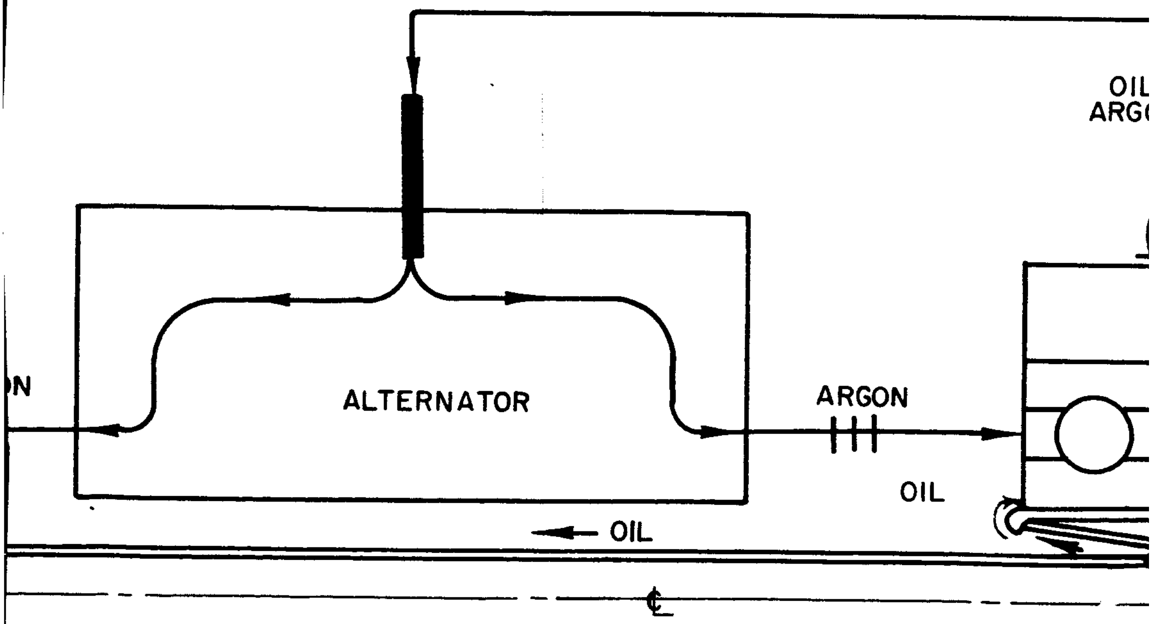


Figure 2 **Preliminary** Schematic of Brayton-Cycle Bearing System





Rolling-Element

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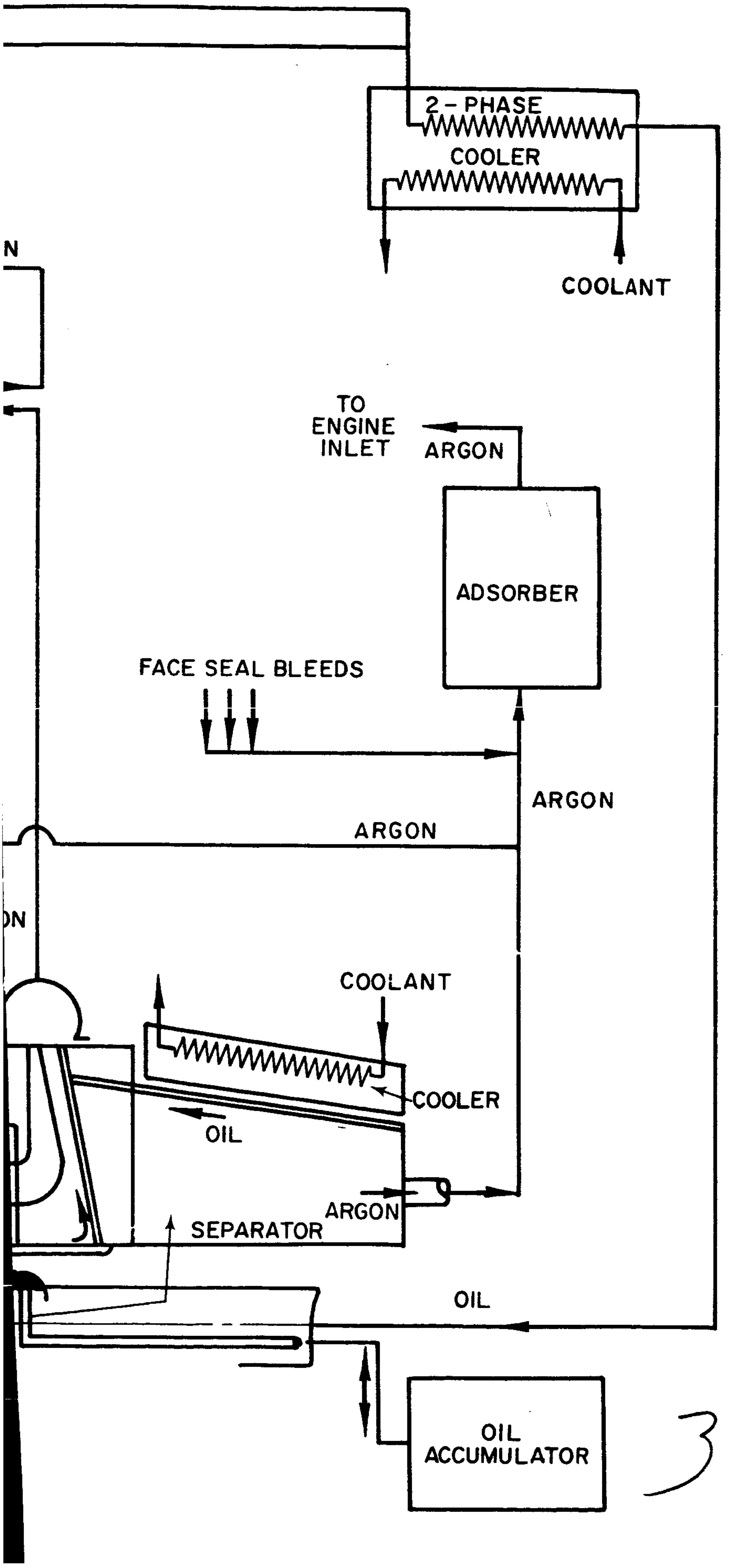

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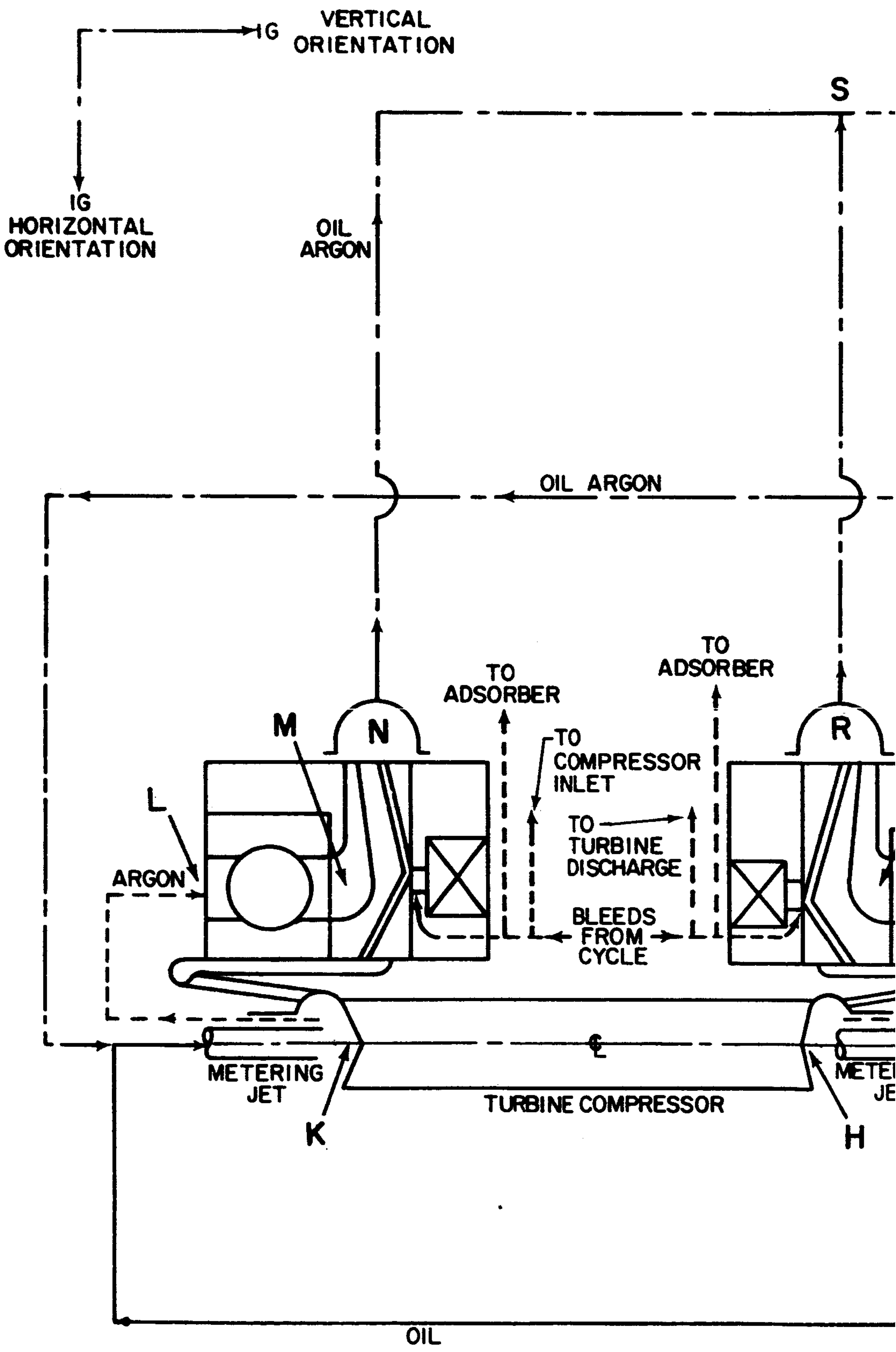
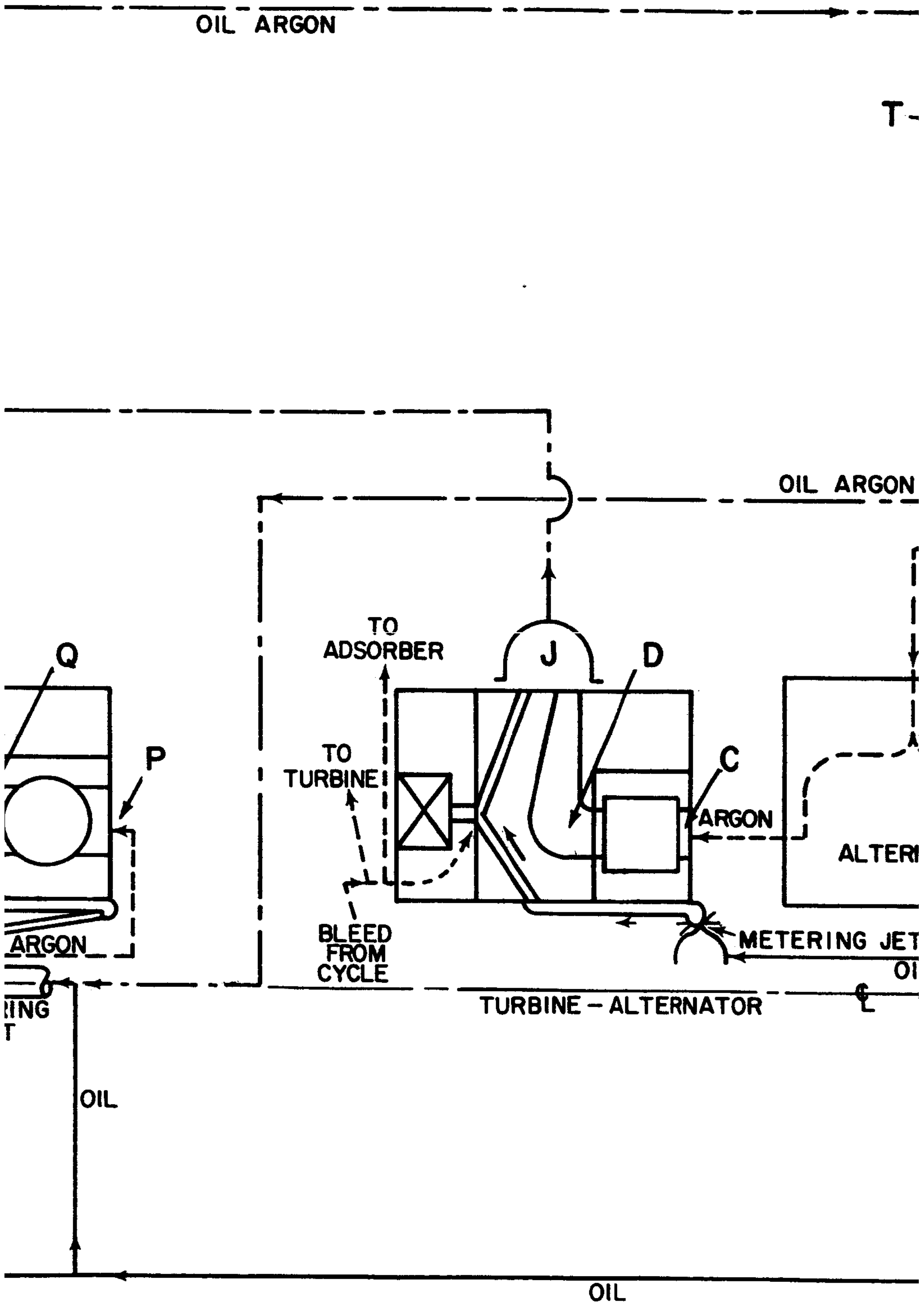
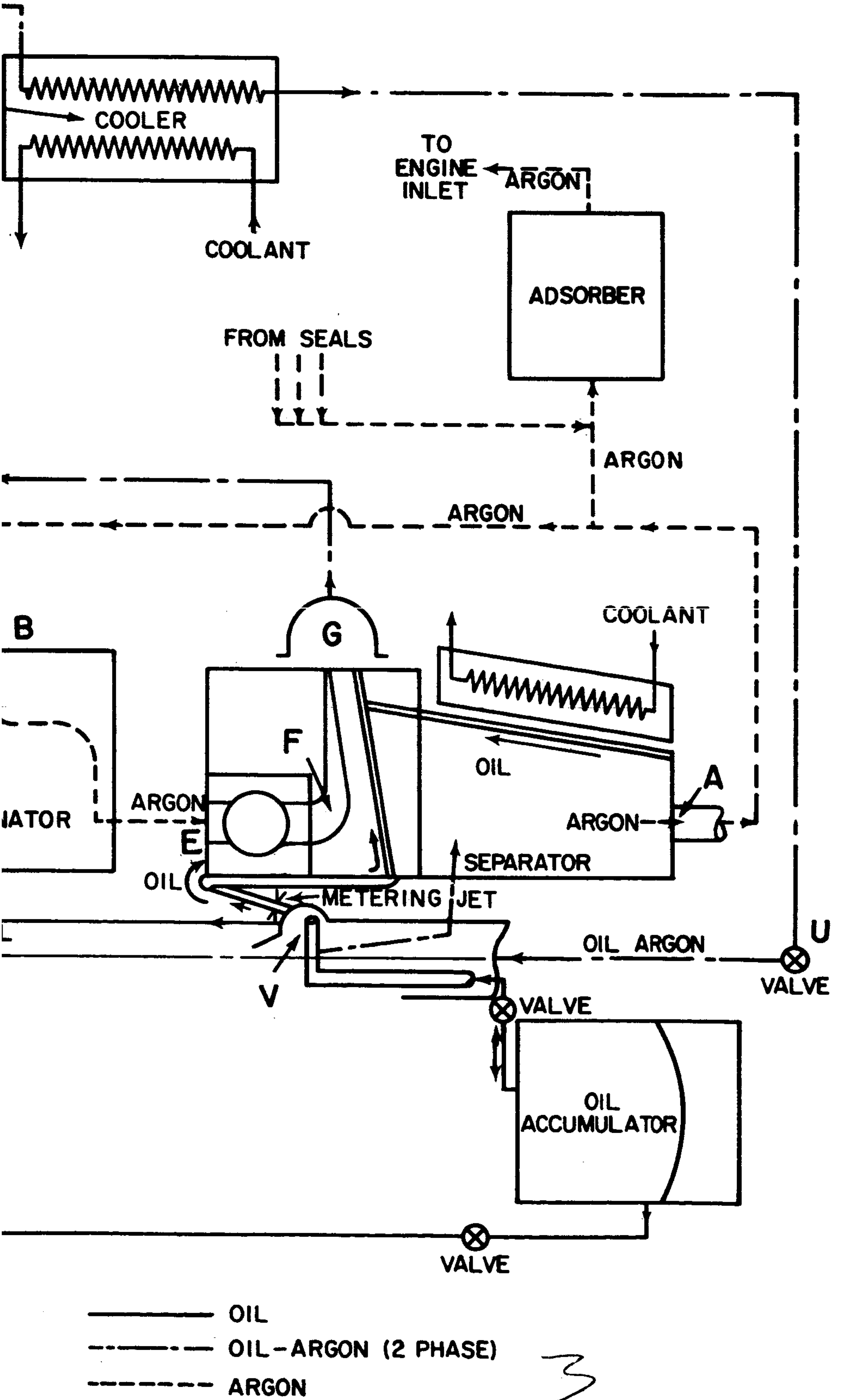


Figure 3 Modified Brayton-Cycle Rolling-Element Bearing S



ystem Schematic



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Total power loss for the revised lubrication system shown in Figure 3 is 1480 watts and a breakdown of these losses is presented in Table 1:

TABLE 1

Brayton-Cycle Rolling-Element Bearing Lubrication System

Power Losses

Type of Loss and Location	Oil Flow, lb /hr	Argon Flow, lb/hr	Power Loss, watts	
Turbine-Compressor Compartment 1				
bearing heat generation	60	-	202.1	
seal heating	60	-	200.0	
oil pumping	60	-	147.5	
argon pumping	-	4	9.8	
total				559.4
Turbine-Compressor Compartment 2				
bearing heat generation	80	-	203.5	
seal heating	80	-	200.0	
oil pumping	80	-	196.8	
argon pumping	-	4	9.8	
total				610.1
Turboalternator Compartment 1				
bearing heat generation	-	-	73.0	
seal heating	-	-	55.8	
oil pumping	10	-	6.2	
argon pumping	-	4	2.4	
total				137.4
Turboalternator Compartment 2				
bearing heat generation	15	-	108.0	
oil pumping	15	-	10.3	
argon pumping	-	4	2.7	
separator drag	-	-	7.1	
oil probe drag	-	-	29.0	
separator pumping	-	8	16.4	
total				173.5
			Total Losses	1480.4

The face seals require argon bleeds through labyrinth seals for proper pressurization to prevent oil leakage. Also, there will be some gas leakage through the face seals. The summary of the bleed and leakage flows is presented in Table 2.

The revised lubrication system shown on Figure 3 is a recirculating system with a number of single-phase and two-phase (oil and argon) flow passages. The pressure drop as a function of argon flow for each section of the system is presented in Figures 4 through 13 and the letters identifying the various locations in the circuit are shown in Figure 3. The pressure drop from the Number 1 bearing compartment in the turboalternator to the Number 1 bearing area in the turbine-compressor, Stations J to K on Figure 3, is presented as 3 curves in Figure 8. The argon-oil mixture must rise 33 inches vertically from J to K when the power system is mounted vertically in a 1 g field. The pressure loss when operating vertically consists of the sum of the basic friction loss for flow in a horizontal pipe and the equivalent head loss. In horizontal operation on the ground or in a zero-gravity environment, the pressure loss will correspond to the friction pressure loss of Figure 8 which was determined by the Lockhart-Martinelli correlation[2]. In vertical operation, the head loss based on an equivalent fluid density is added to produce the total pressure loss in Figure 8. The flow from the turboalternator Number 2 bearing area to the Number 2 compartment in the turbine-compressor, Stations G to H, involves a vertical rise of about 22 inches and the pressure loss again is dependent upon orientation, as indicated in Figure 7. Another vertical two-phase upward flow occurs in the return line from the bottom of the turboalternator to the separator. A vertical rise of 11.2 inches is involved. The pressure loss for this section, Stations U to V, is presented in Figure 12 in a manner similar to Figure 8.

The system also includes two-phase flow in the downward direction when the powerplant is oriented vertically. This flow passes through an oil cooler which consists of a single-jacketed pipe with liquid coolant circulated in the jacket. The estimated pressure losses for both vertical and horizontal operation for this return of the two-phase flow are presented in Figures 10 and 11.

The scavenge pumps located on the turbine-compressor shaft (integral with the seal plates) have a tip velocity of about 460 feet/second. Assuming the oil has no influence on the pump performance, the static pressure rise across the impeller is about 0.46 psi. The losses in the exit volute and diffuser could be in the range of 0.34 to 0.56 psi. Therefore, the conservative assumption was made that the kinetic energy at the discharge

TABLE 2

Brayton-Cycle Rolling-Element Bearing Lubrication System

Gas Leakage and Bleed Flows

Seal Location	Type	P_{up}-stream psia	P_{down}-stream psia	Temp., °F	Leakage, lb/hr	Per Cent of Cycle Flow
Turbine-Compressor						
front compressor seal	labyrinth	11.0	5.5	340	18.2	0.843
rear compressor front turbine	labyrinth	12.83	9.26	340	20.7	0.958
rear turbine seal	labyrinth	11.0	8.29	340	13.3	0.616
No. 1 bearing seal	carbon (wet face)	9.90	-	340	0.31	0.0144
No. 2 bearing seal	carbon (wet face)	10.0	-	340	0.313	0.0145
No. 1 compart. capillary bleed	capillary		6.0		1.4	0.065
No. 2 compart. capillary bleed	capillary		6.0		1.4	0.065
Turboalternator						
rear turbine seal	labyrinth	11.0	6.73	340	25.8	1.19
bearing seal	carbon (wet face)	11.0	-	340	0.62	0.0287
No. 1 compart. capillary bleed	capillary	11.0	6.0	340	1.4	0.185
metered flow to front of turbine	capillary	11.0	-	340	4.00	0.185
Total					87.443	4.0446

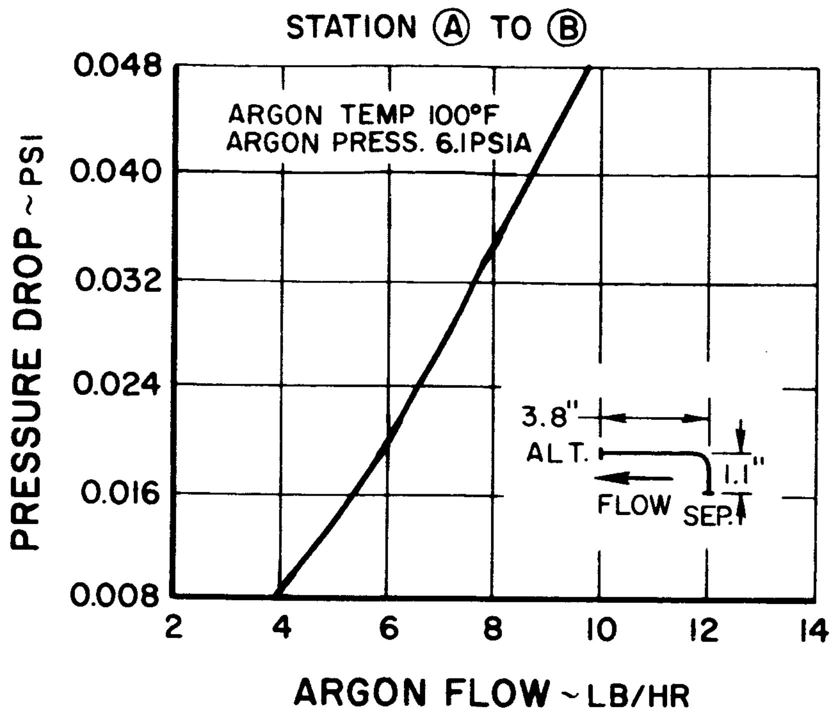


Figure   Pressure Drop for Tubing from Separator to Alternator

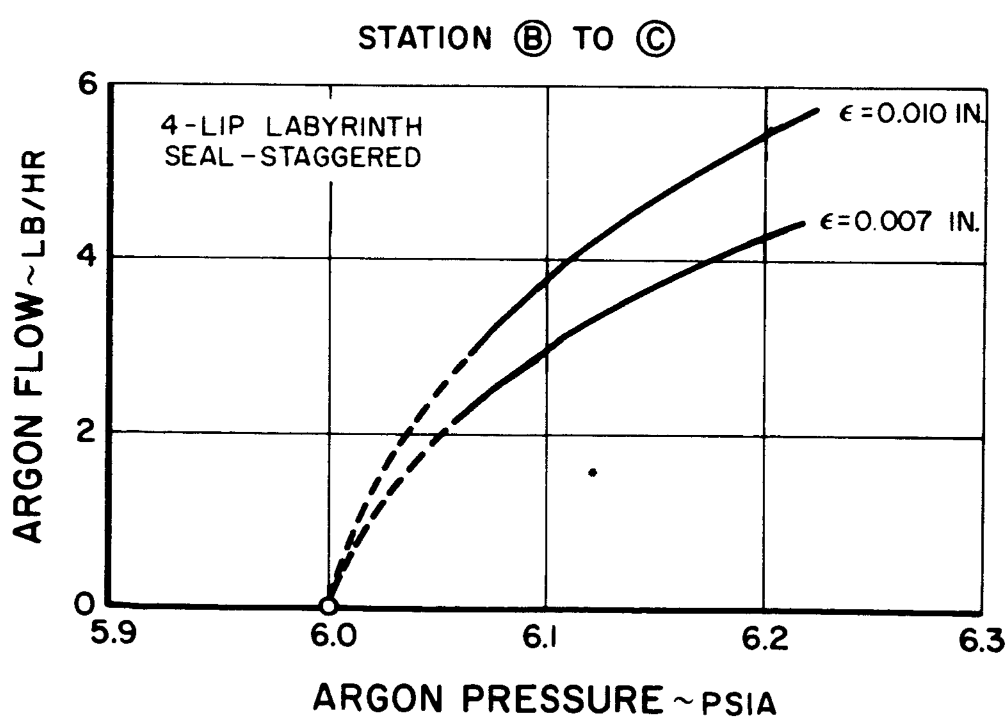


Figure   Turboalternator Labyrinth Seal Flow. Alternator Cavity Purge

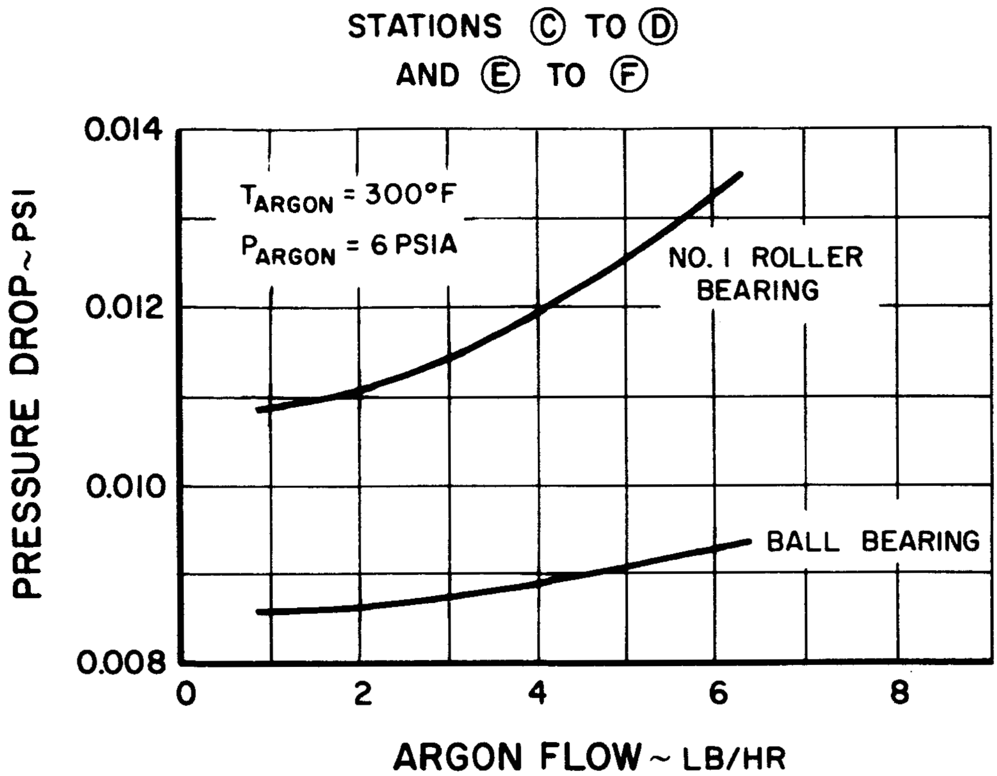


Figure 6 Turboalternator Bearing Pressure Drop vs Gas Flow Rate

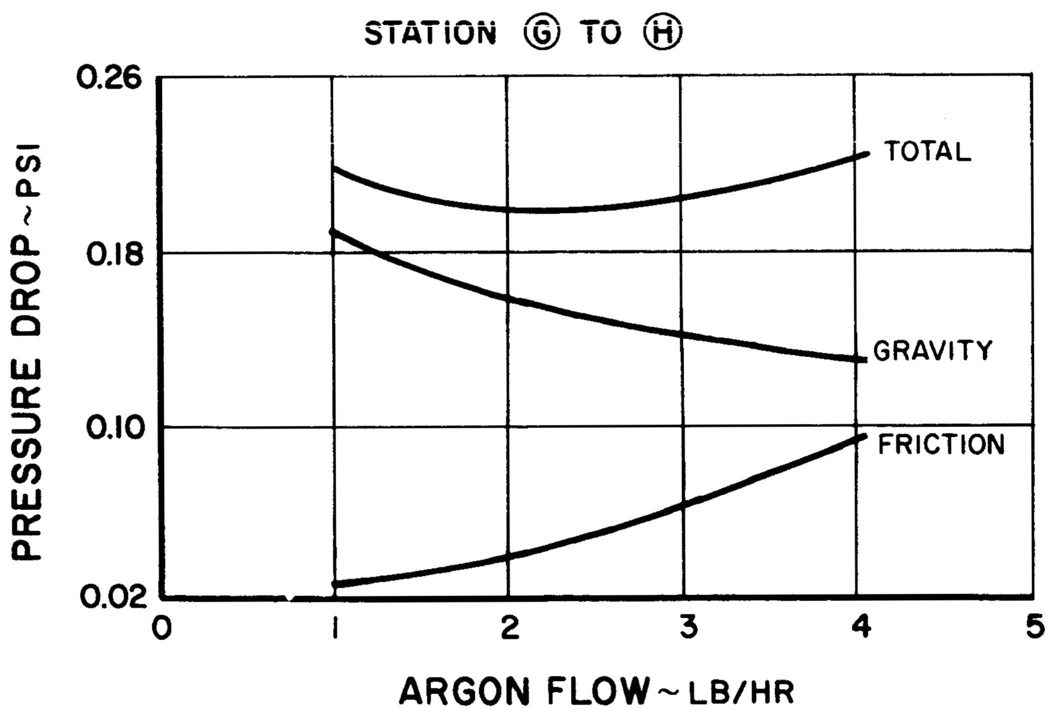


Figure 7 Pressure Drop for Tubing from Turboalternator No. 2 Area to Turbine-Compressor No. 2 Area.

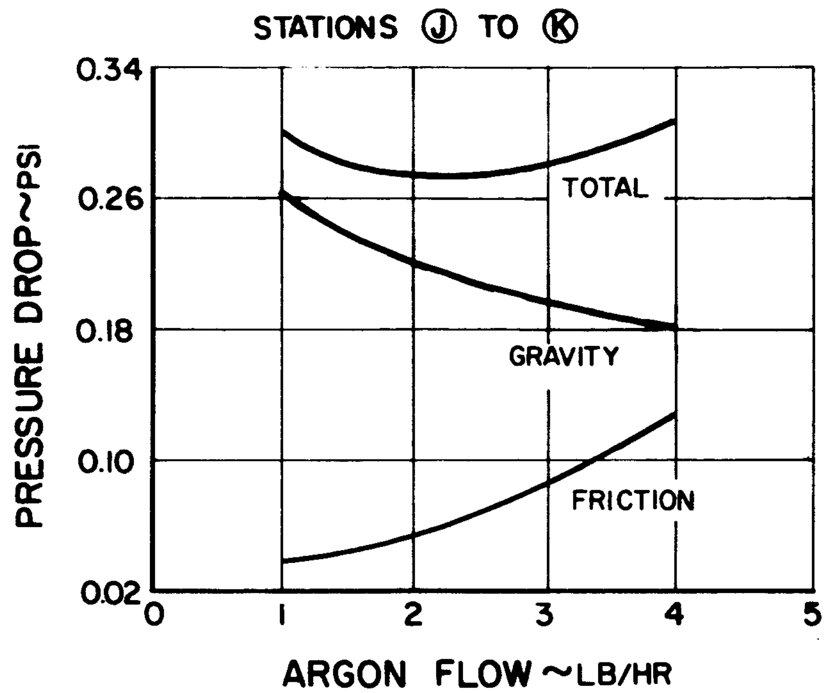


Figure 8 Pressure Drop for Tubing from Turboalternator No. 1 Area to Turbine-Compressor No. 1 Area.

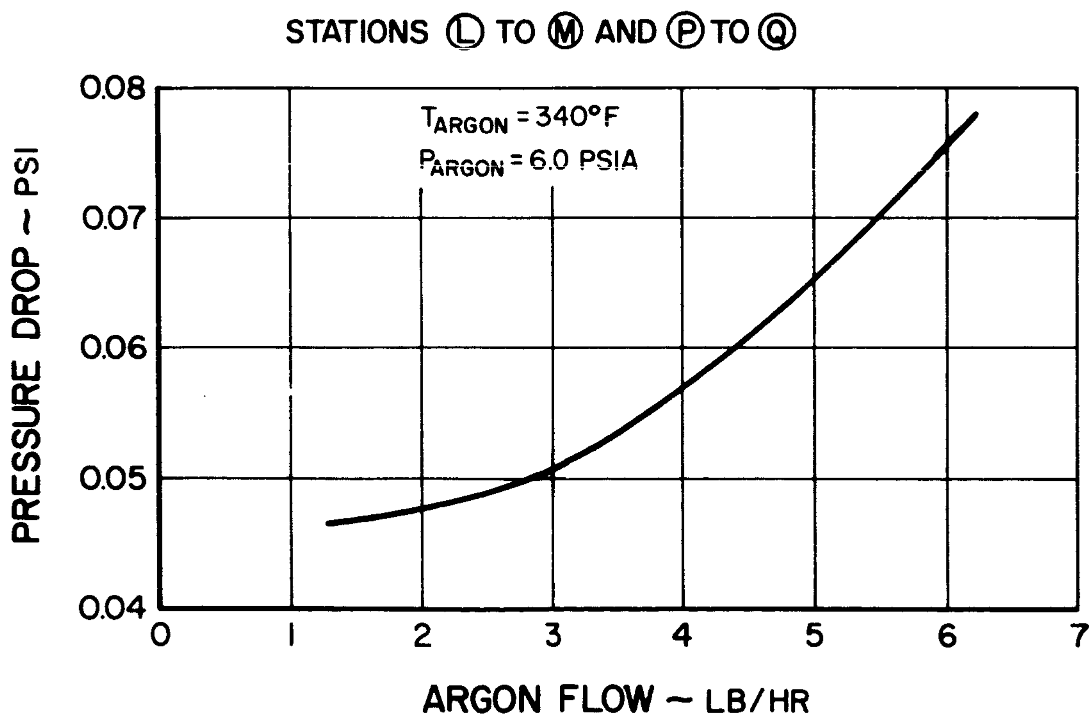


Figure 9 Turbine-Compressor Bearing Pressure Drop vs Gas Flow

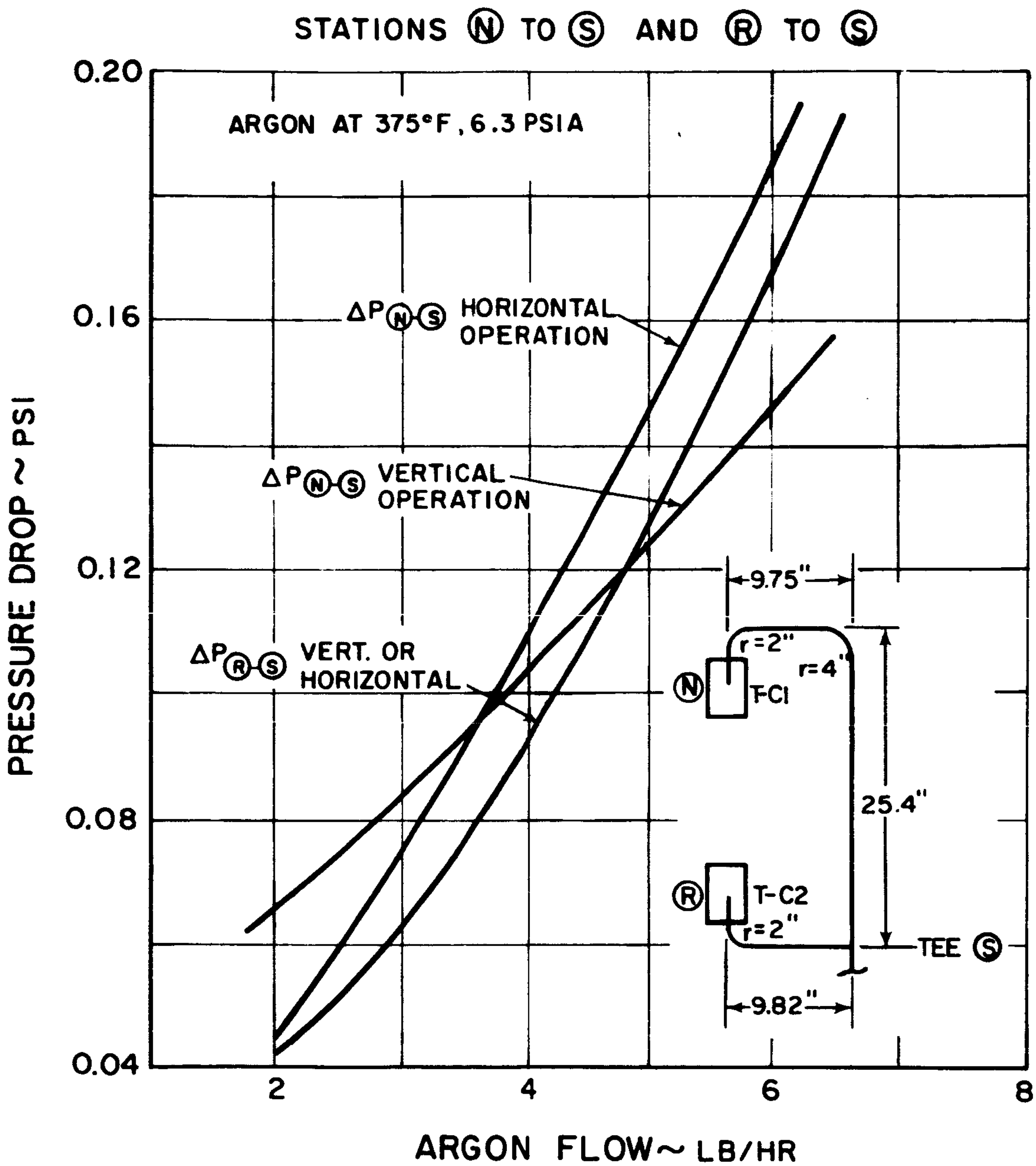


Figure 10 Pressure Drop for Tubing from Turbine-Compressor to Tee Upstream of Cooler

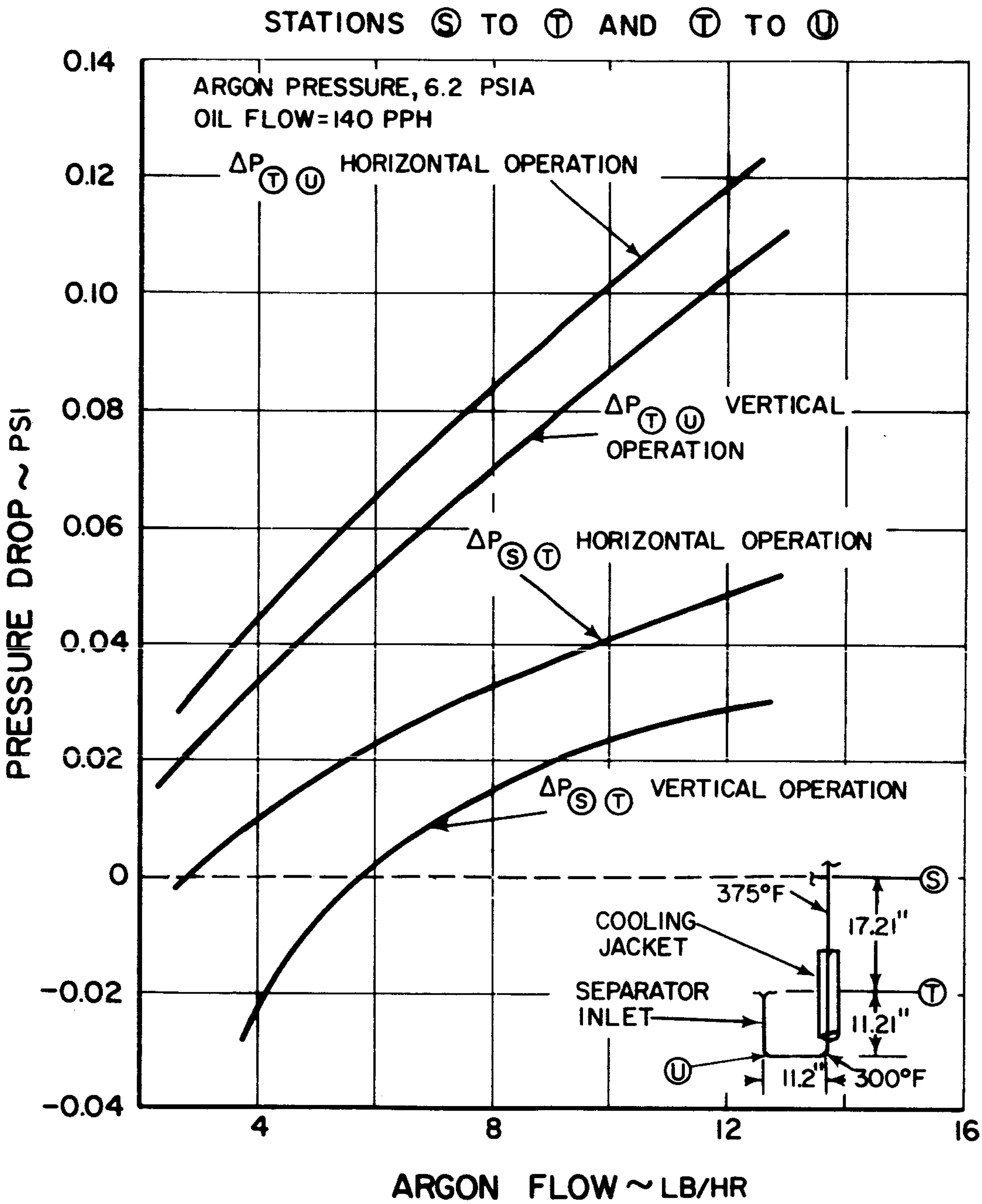


Figure 11 Pressure Drop for Tubing from Tee to Mid-Cooler, and Mid-Cooler to Separator Riser

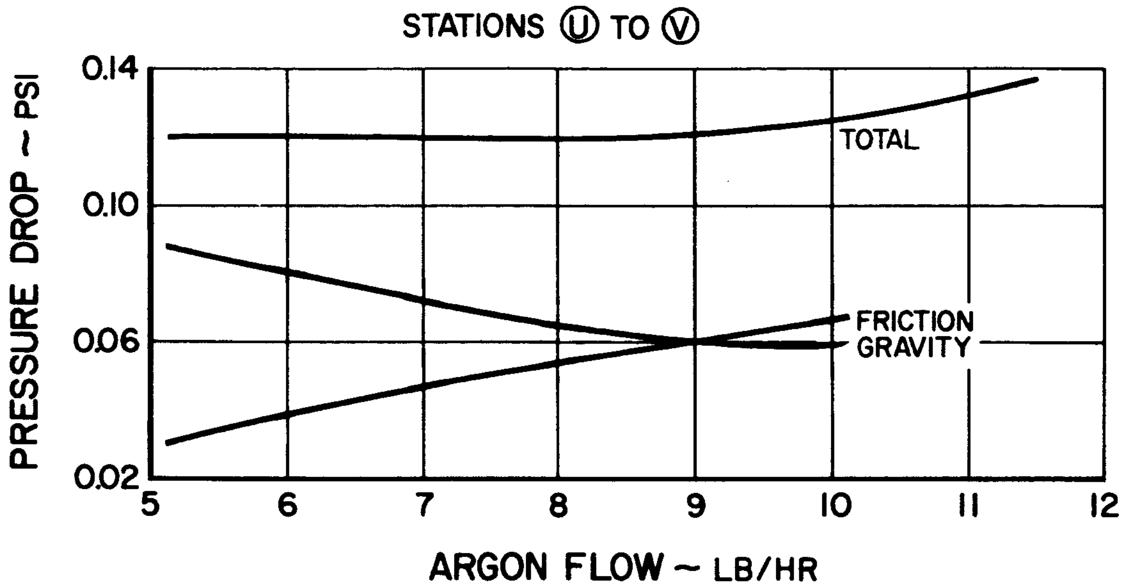


Figure 12 Vertical Lift into Separator

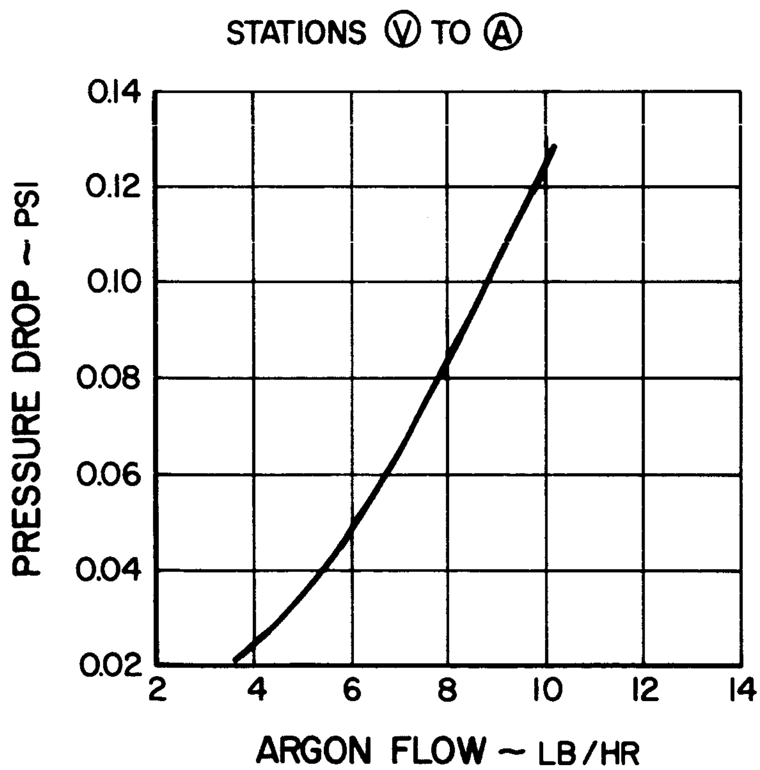


Figure 13 Pressure Drop in Separator

of the impeller is lost. The scavenge pumps in the turboalternator were evaluated in a similar manner. The separator contains a small impeller capable of providing some further pressure rise which just about balances the pressure losses in the separator. The estimated sum of the pressure increases provided by the scavenge pumps is 0.66 psi. Matching the various pressure losses in the system with this driving pressure indicates a system flow of 8.55 pounds per hour of argon into the separator area for operation in the horizontal or zero gravity environment. In the vertical direction, the corresponding argon flow is 6.10 pounds per hour. A summary of the zero gravity or horizontal operating conditions estimated for the lubrication system in Figure 3 is presented in Table 3. The corresponding conditions for vertical operation on the ground with the compressor end up are presented in Table 4.

TABLE 3

Brayton-Cycle Rolling-Element Bearing Lubrication System

Summary of Pressure Losses

Horizontal or Zero-Gravity Operation

Station on Figure 3	Description	Oil Flow, lb/hr	Argon Flow lb/hr	Press. Loss, psi
A-B	line from separator to alternator	-	6.75	0.024
B-C	labyrinth seal	-	3.17	0.073
C-D	roller bearing	-	3.17	0.090
J-K	line from turboalternator to turbine-compressor	10	3.77	0.116
L-M	ball bearing	10	3.77	0.055
N-S	scavenge line	60	4.07	0.111
B-E	labyrinth seal	-	3.58	0.090
E-F	ball bearing	-	3.58	0.009
G-H	line from turboalternator to turbine-compressor	15	4.18	0.102
P-Q	ball bearing	15	4.18	0.058
R-S	scavenge line	80	4.48	0.109
S-T	line to cooler	140	8.55	0.035
T-U	line from cooler	140	8.55	0.089
U-V	scavenge return	140	8.55	0.058
V-A	separator	4	8.55	0.094

TABLE 4

Brayton-Cycle Rolling-Element Bearing Lubrication System

Summary of Pressure Losses

Vertical Operation

Station on Figure 3	Description	Oil Flow lb/hr	Argon Flow lb/hr	Press. Loss, psi
A-B	line from separator to alternator	-	4.30	0.009
B-C	labyrinth seal	-	1.40	0.020
C-D	roller bearing	-	1.40	0.011
J-K	line from turboalternator to turbine-compressor	10	2.00	0.273
L-M	ball bearing	10	2.00	0.048
N-S	scavenge line	60	2.30	0.072
B-E	labyrinth seal	-	2.90	0.062
E-F	ball bearing	-	2.90	0.009
G-H	line from turboalternator to turbine-compressor	15	3.50	0.214
P-Q	ball bearing	15	3.50	0.054
R-S	scavenge line	80	3.80	0.086
S-T	line to cooler	140	6.10	0.002
T-U	line from cooler	140	6.10	0.053
U-V	scavenge return	140	6.10	0.223
V-A	separator	2	6.10	0.050

An examination of the two-phase flow conditions indicates that annular flow can be expected when the system is operating horizontally or in zero gravity. Ovid Baker[3] has developed a correlation of various two-phase flow phenomena in terms of a liquid and a gas flow parameter for flow in horizontal pipes. This correlation is presented in Figure 14 with the different flow regimes marked. Actually, the demarcation between the flow regimes is not sharp, but rather the lines in Figure 14 represent transition regions. The flow conditions for the various two-phase sections in the lubrication system for horizontal operation are included in Figure 14. These points all fall in the annular flow regimes although some are close to slug flow (the line from the Number 1 turboalternator cavity to the Number 1 turbine-compressor area and the Number 1 turbine-compressor scavenge line).

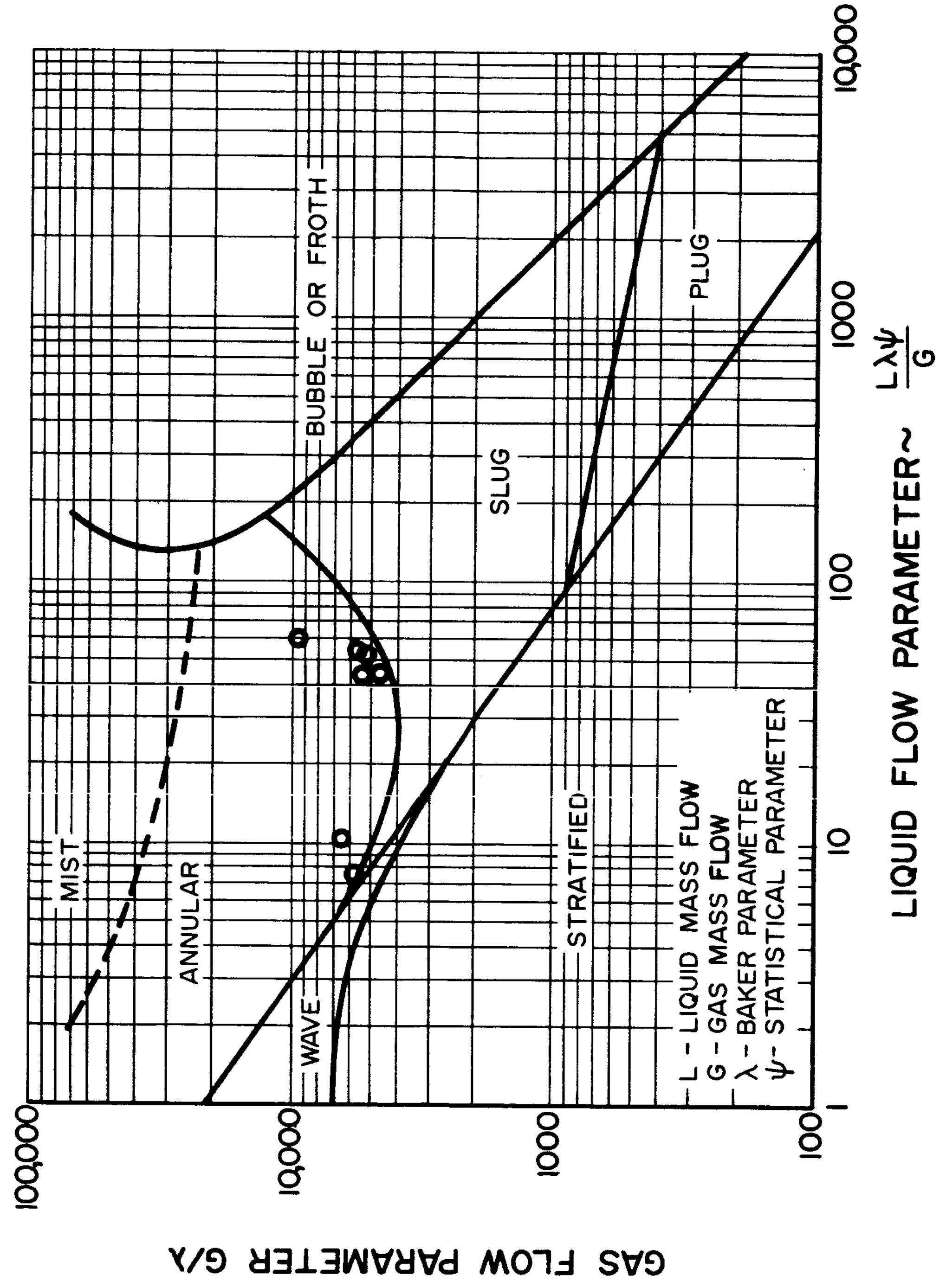


Figure 14 Flow Regime Chart. Horizontal Orientation

In vertical operation Griffith and Wallis[4] have developed a two-phase flow regime correlation based on the limited vertical flow data available. This correlation is presented in Figure 15 and the three major vertical flows in the lubrication system are included in this plot. Figure 15 indicates that the vertical flows are all well within the annular or mist flow regime. There is some question about this result. The Baker correlation is for horizontal flow only, but if these vertical flow conditions were placed on the Baker correlation they would fall in the slug-flow regime. Two-phase flow is not a well-defined art and evidently requires experimental verification. References 5 through 18 are further discussions of two-phase flow phenomena.

If higher argon flows were used in the system, all of the two-phase flows, both vertical and horizontal, could fall in the annular regime on the Baker correlation. In order to achieve these higher flows higher pressure rise in the scavenge pumps would be required. Figure 16 presents the overall pressure losses which must be matched by the scavenge pump pressure as a function of argon flow. Actually, the pressure rise in the pumps as designed is difficult to establish. The assumption was made that the kinetic energy added to the oil was lost and made no contribution to the pressure rise. The kinetic energy added to the oil in the scavenge pumps is many times the energy added to the argon, as indicated in Table 1. If it is a conservative assumption that the energy in the oil is lost, a higher pressure rise may be achieved. Therefore, the lubrication system may operate with increased margin from slug flow over the situation presented in Figure 14.

One alternate approach is to use a scavenge pump designed to utilize the kinetic energy in the oil rather than in the argon. Since this kinetic energy is 5 to 20 times that of the argon, significant improvements in pressure rise may be possible. Since the oil is rotated in the various grooves for cooling, no additional power loss is imposed on the system when the kinetic energy of the oil is used for pumping.

An alternate scavenge pump design, Figure 17, is being developed based on the jet pump principle where the oil is the energizing fluid and the argon is the medium being pumped. The conventional configuration of a single jet in a circular pipe is not practical for this machinery. The primary jet is replaced by a sheet of oil discharged radially from a rotating impeller. This stream of oil enters a narrow annular passage which serves as the throat of the pump. Since the passage is radial it also serves as a diffuser. The high velocity oil drags argon from the surrounding cavity into the passage, where the two fluids mix and exchange energy. This annular form of the jet pump has the inherent

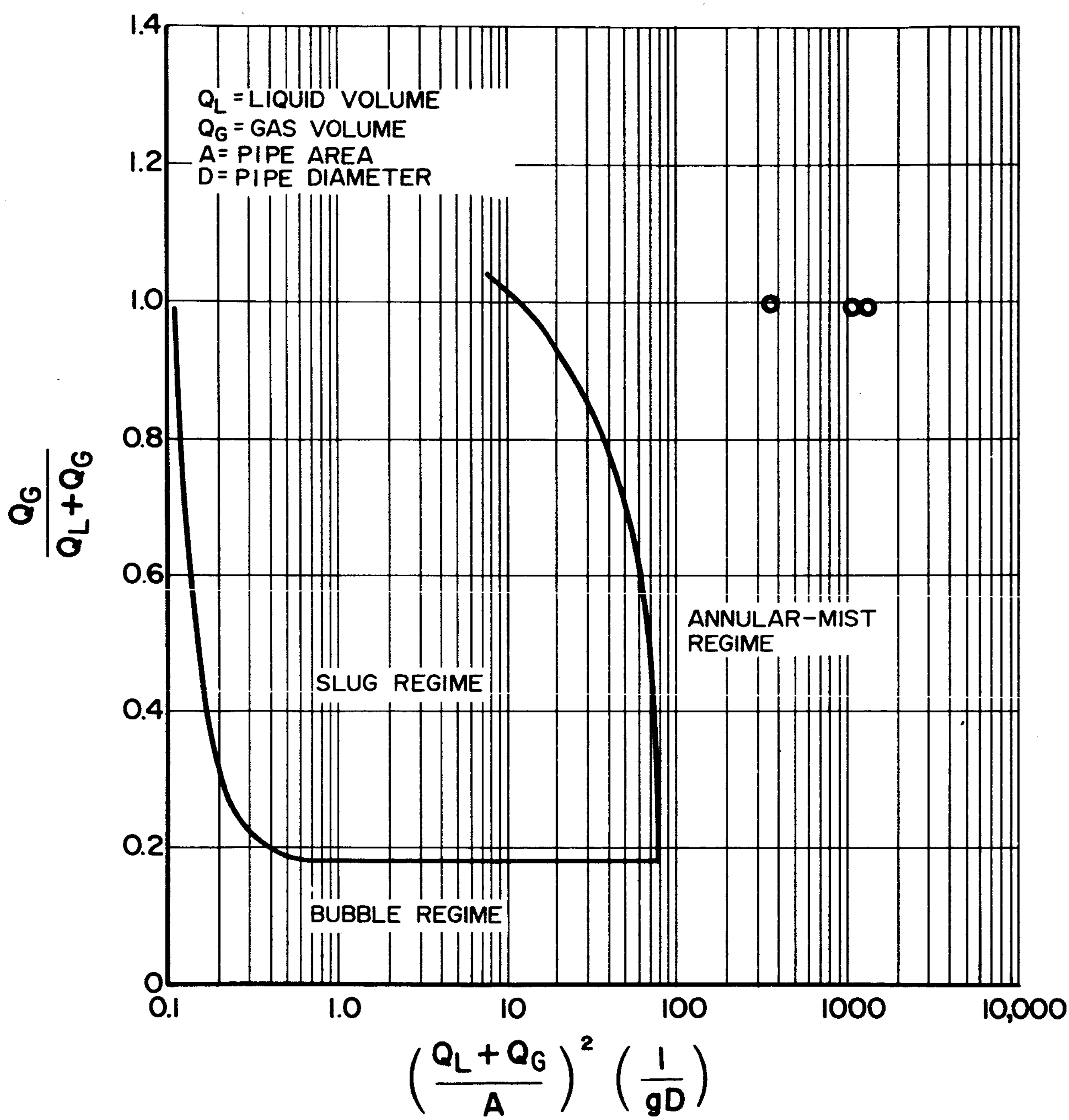


Figure 15 Flow Regime Chart. Vertical Operation

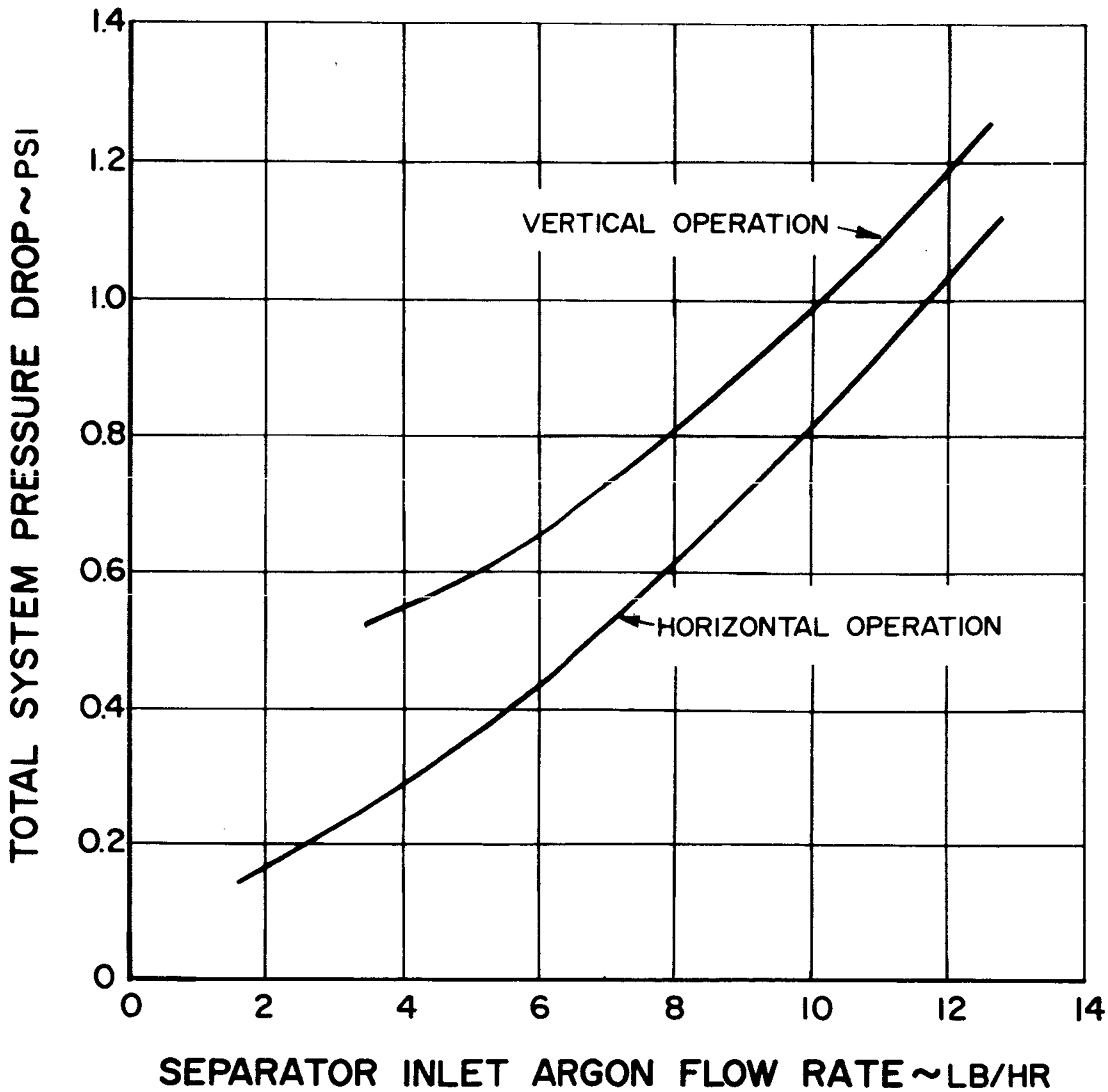


Figure 16 System Pressure Drop and Flow Characteristics

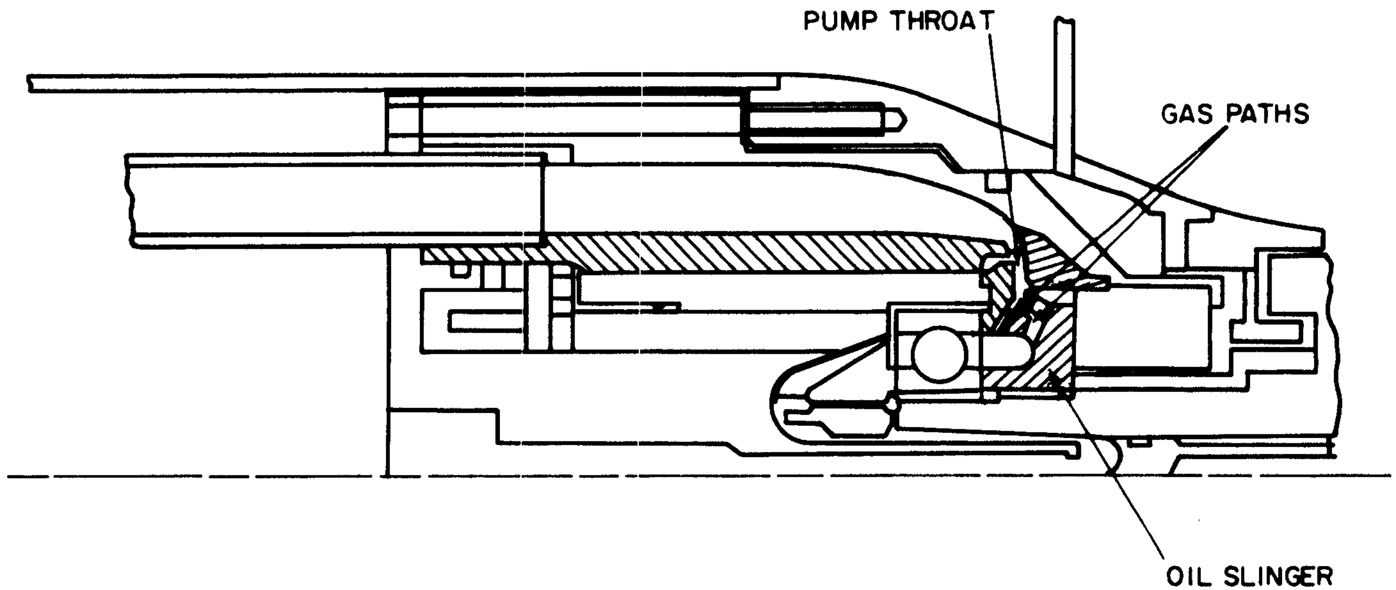


Figure 17 Turbine-Compressor Bearing Compartment Showing Annular Jet Scavenge Pump Concept

advantage of providing a large interface area between the primary and secondary fluids in a short throat section. The performance potential of the annular oil jet pump is very high with a pressure rise of between 1 and 5 psi, depending primarily on the width of the annular passage. Figure 18 presents the ideal pressure rise as a function of throat width. Pump performance as a function of argon flow is presented in Figures 19, 20, and 21 for various throat widths. Since the oil sheet must enter the annular passage without hitting the walls of the passage to realize proper fluid mixing, the performance is limited by the annular passage size that can be used within the limits of the mechanical arrangement. In the turbine-compressor scavenge pump a passage width between 0.045 and 0.090 inch appears practical.

If the annular jet pump were to perform as predicted in Figure 21 with a passage width of 0.090 inch (a conservative value) in the turbine-compressor, the pump pressure rise would be over 1.0 psi,providing a system pressure rise of about 1.2 psi. In the horizontal or zero-gravity environment, the argon flow into the separator area would be 12.9 pounds per hour. In the vertical position the corresponding flow would be about 12.1 pounds per hour. A summary of the operating conditions in the horizontal case is listed in Table 5 and the vertical operating conditions are presented in Table 6. The two-phase flow conditions for the improved scavenge pump system are in the annular regime as shown on the Baker correlation in Figure 22 and the Griffith and Wallis correlation in Figure 23. Actually, when the Baker correlation for horizontal flow is applied to vertical operation, annular flows are also indicated.

Another method of utilizing the kinetic energy imported to the oil is to scoop the oil from a rotating pool. High oil pressures can be developed in this manner, but generally at some sacrifice in losses. Such an alternate is presented schematically in Figure 24. In this case, the only two-phase vertical rise occurs from Stations U to V in Figure 24, the return line to the separator area. All of the oil returning from the turbine-compressor passes through the turboalternator. The small scoop in the Number 2 compartment of the turboalternator is replaced by a flow divider which directs the proper amount of oil to each bearing location. The oil leaving the turboalternator is not carried by argon gas to the turbine-compressor but rather it is scooped and returned to the accumulator. Argon with a small quantity of oil mist or vapor flows from the turboalternator to the turbine-compressor.

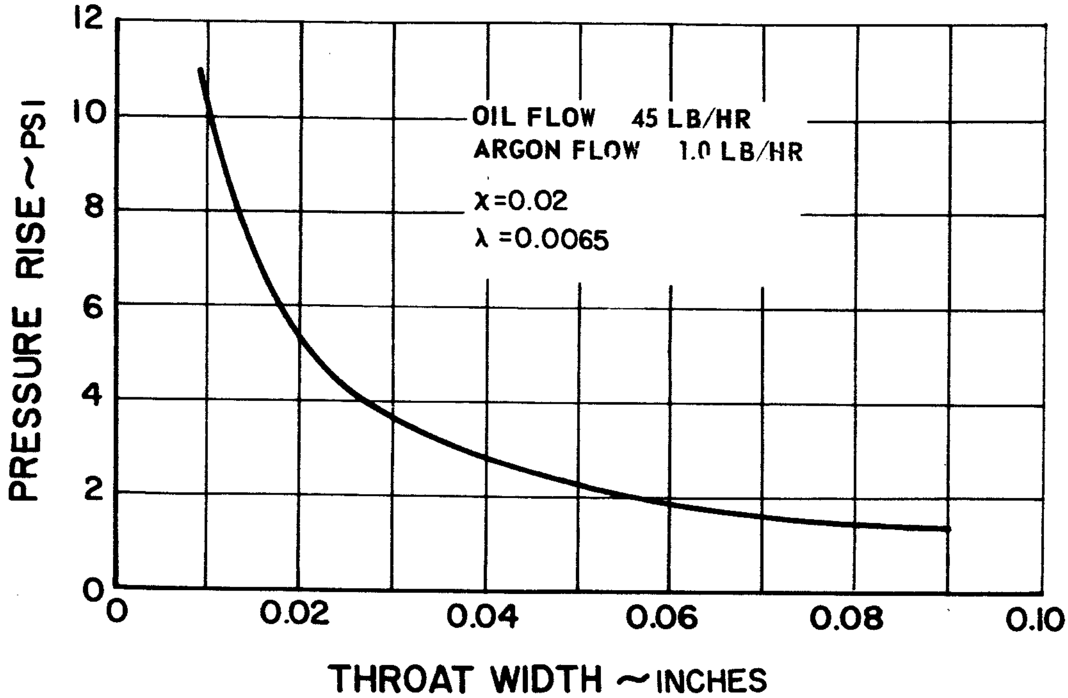


Figure 18 Annular Jet Scavenge Pump for Turbine-Compressor. Effect of Throat Width on Pressure Rise

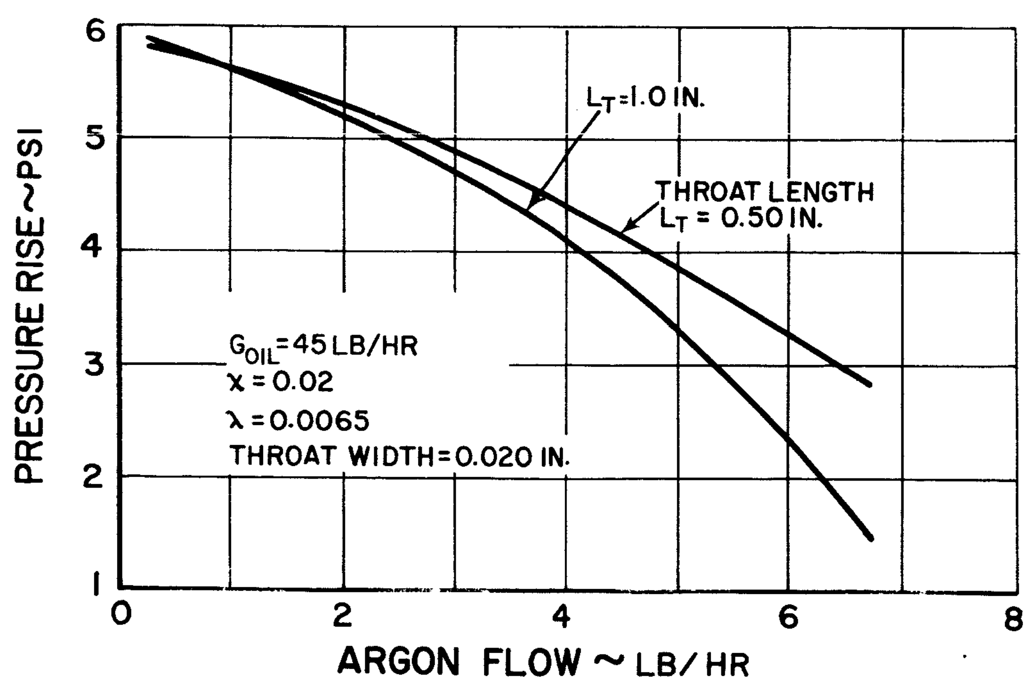


Figure 19 Pressure-Flow Characteristics of Annular Jet Scavenge Pump for Turbine-Compressor

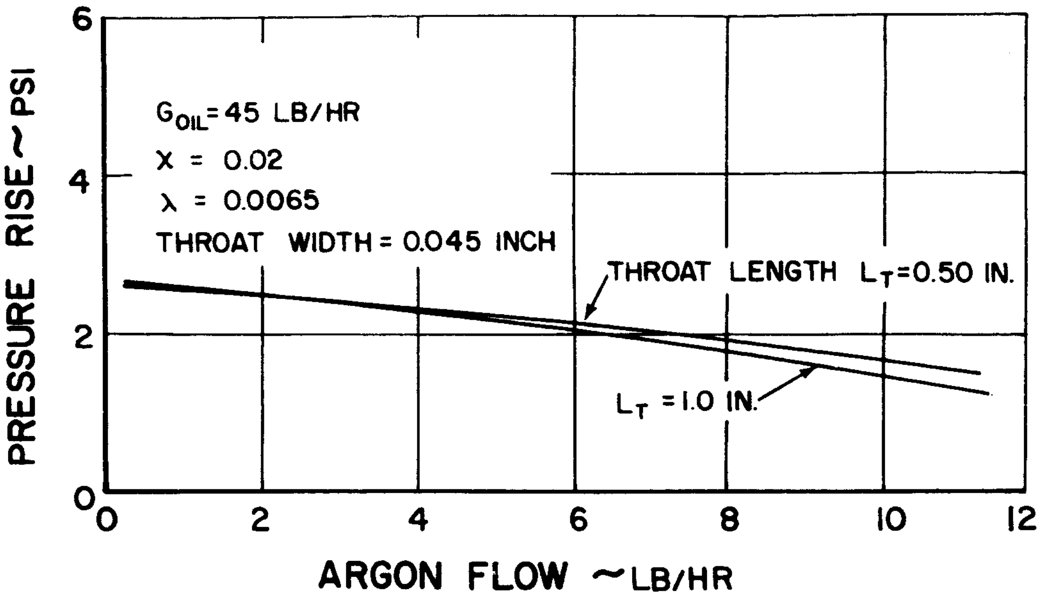


Figure 20 Pressure-Flow Characteristics of Annular Jet Scavenge Pump for Turbine-Compressor

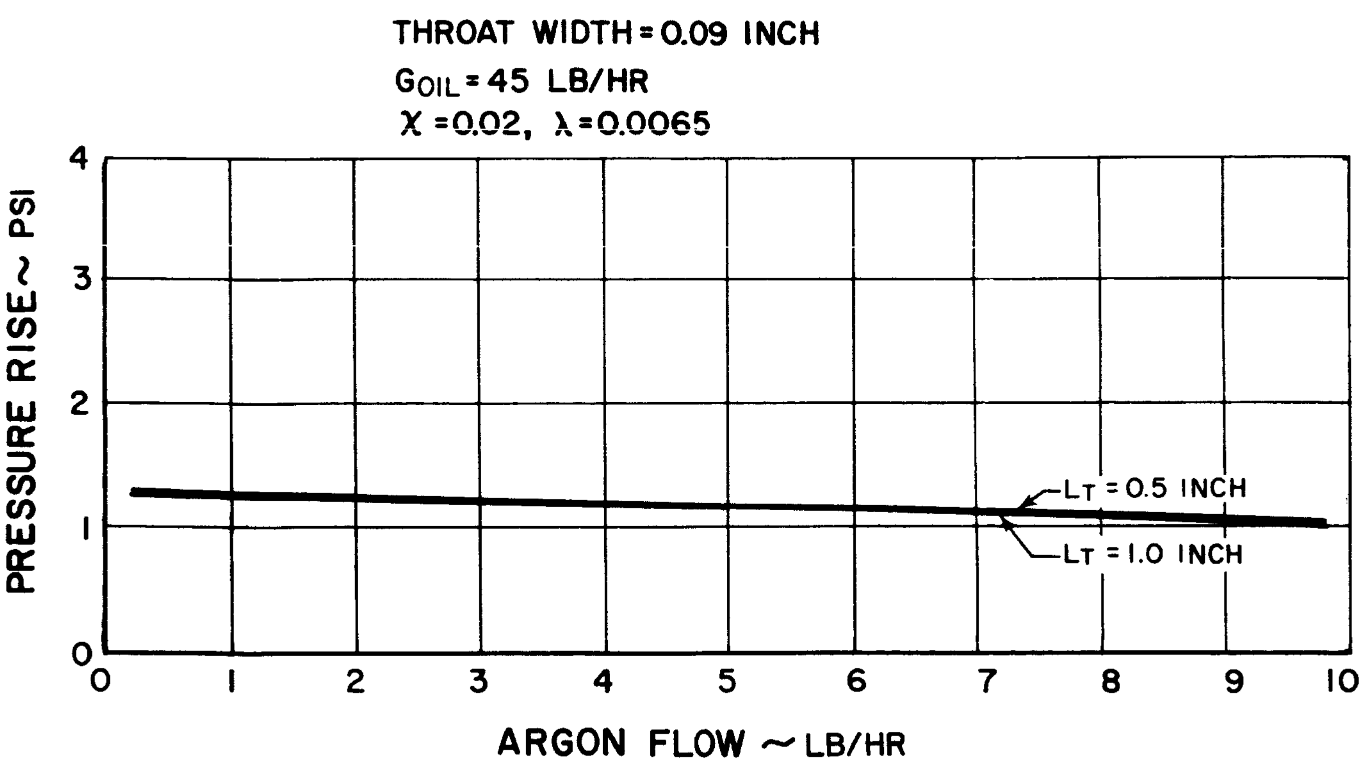


Figure 21 Pressure-Flow Characteristics of Annular Jet Scavenge Pump for Turbine-Compressor

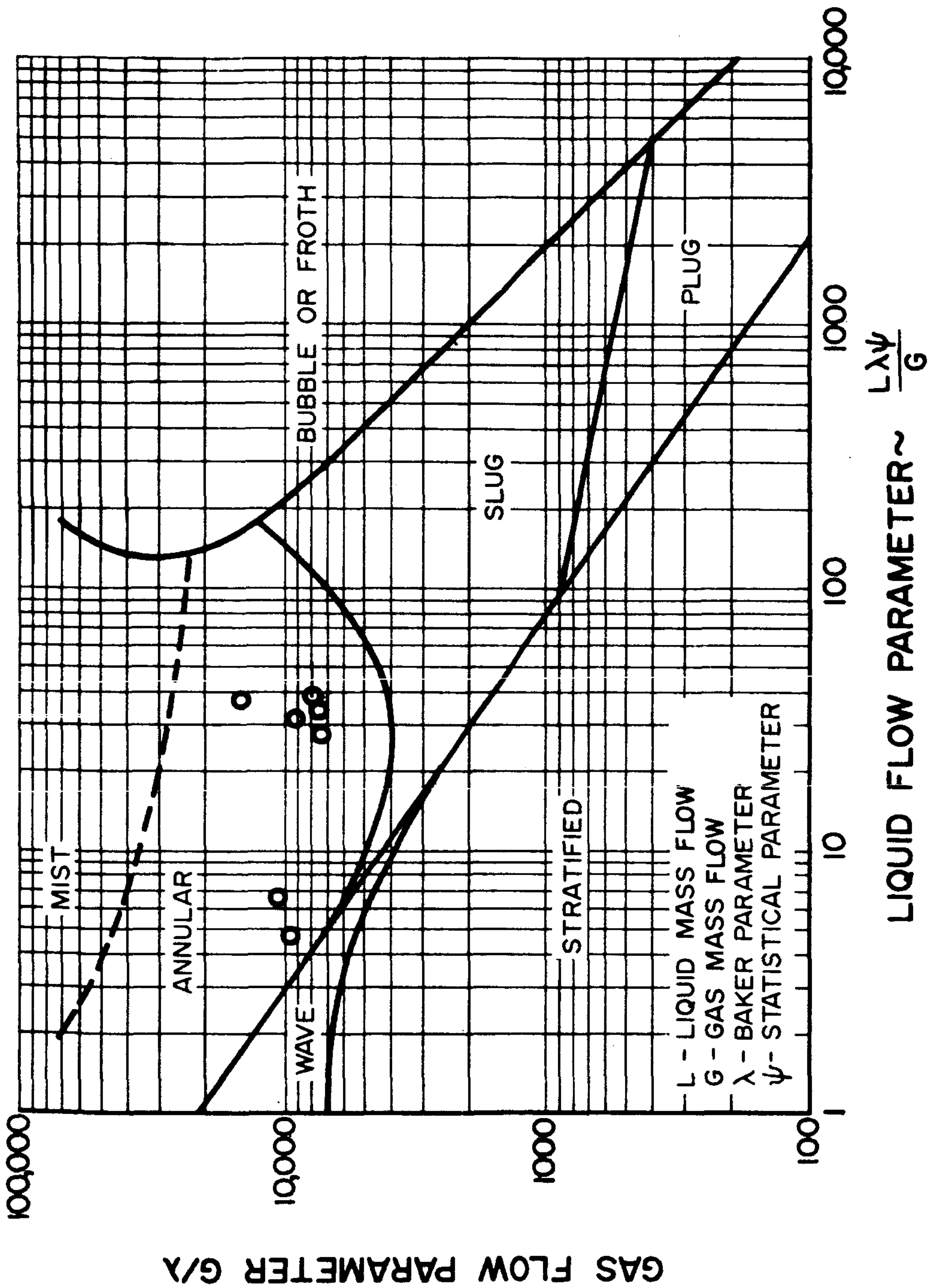


Figure 22 Flow Regime Chart. Horizontal Orientation

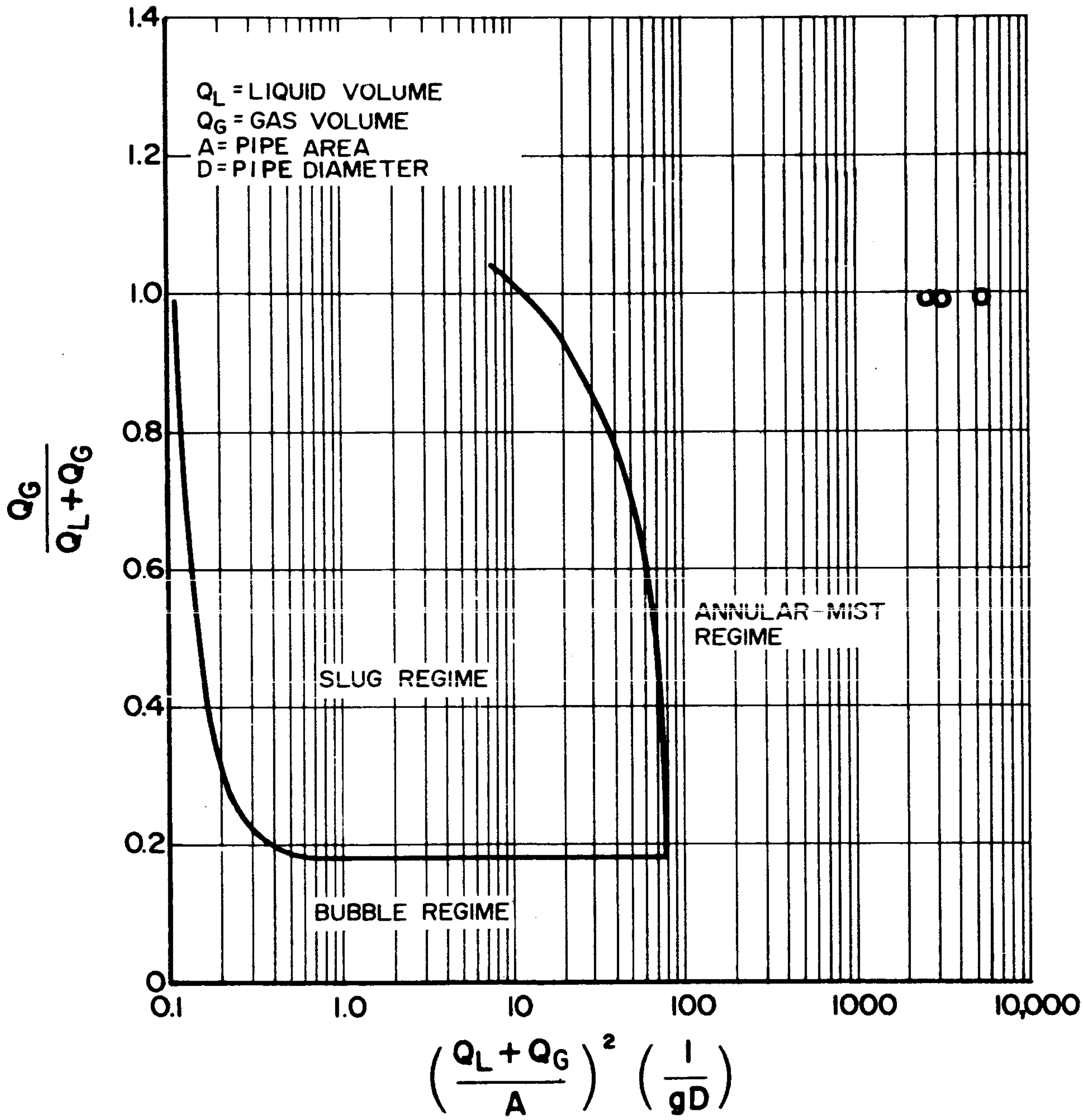


Figure 23 Flow Regime Chart. Vertical Operation

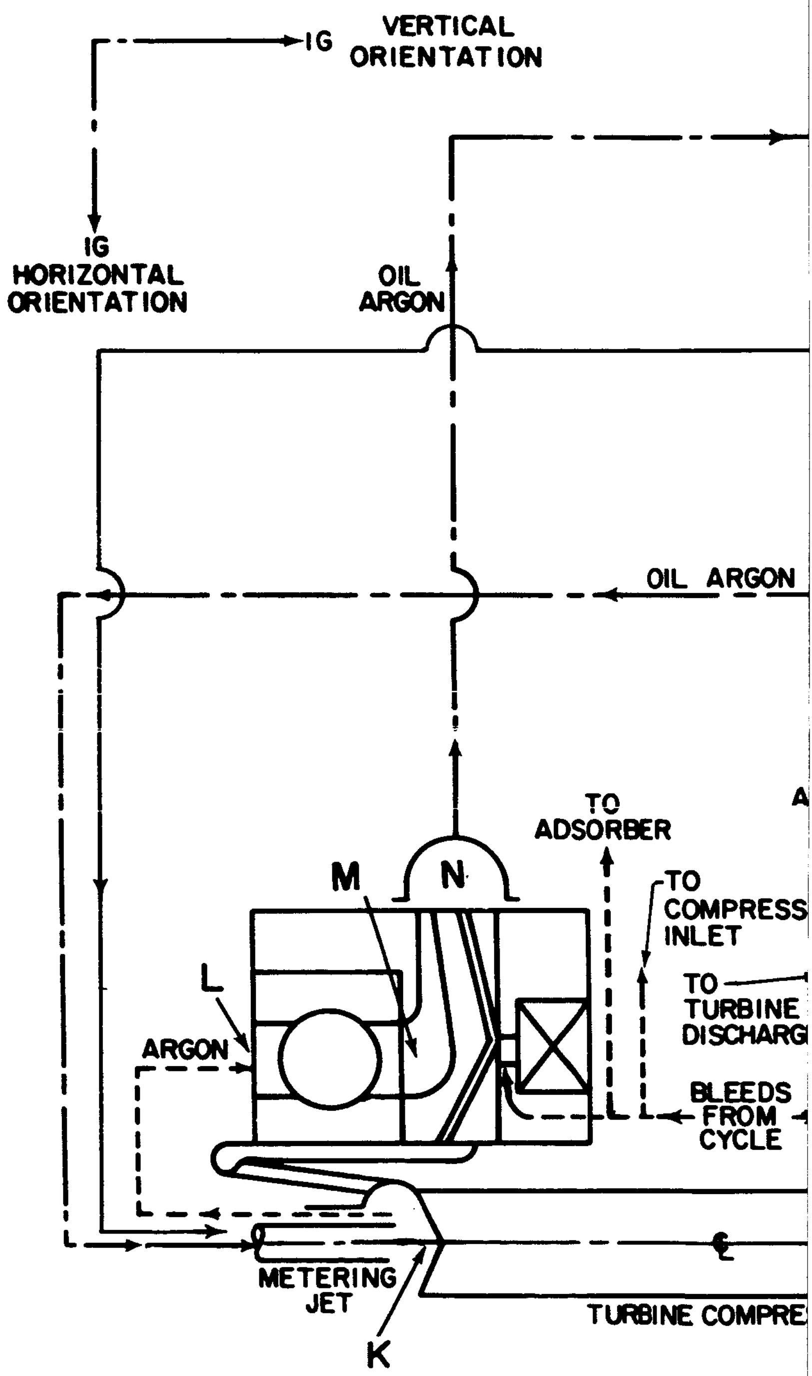
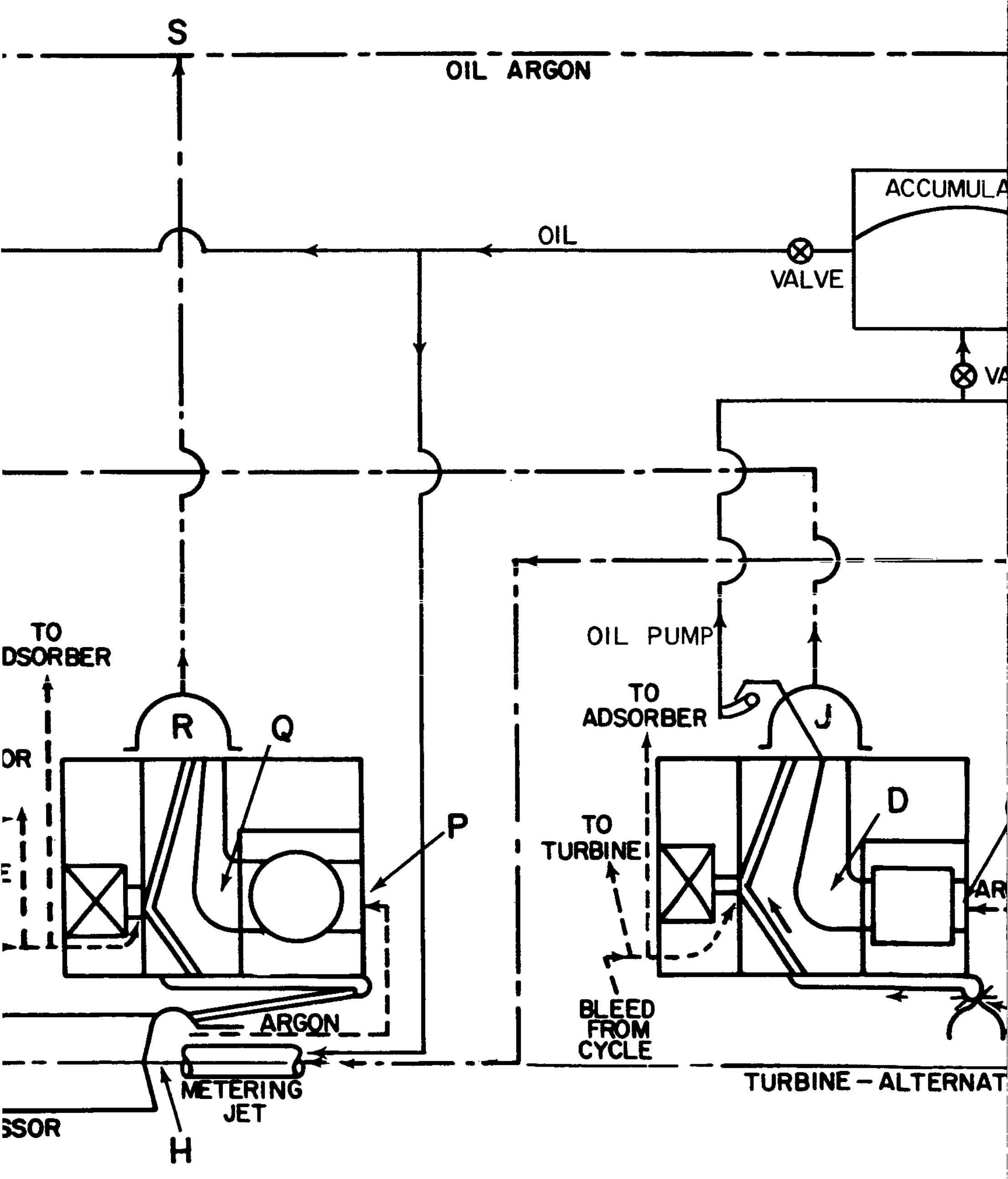
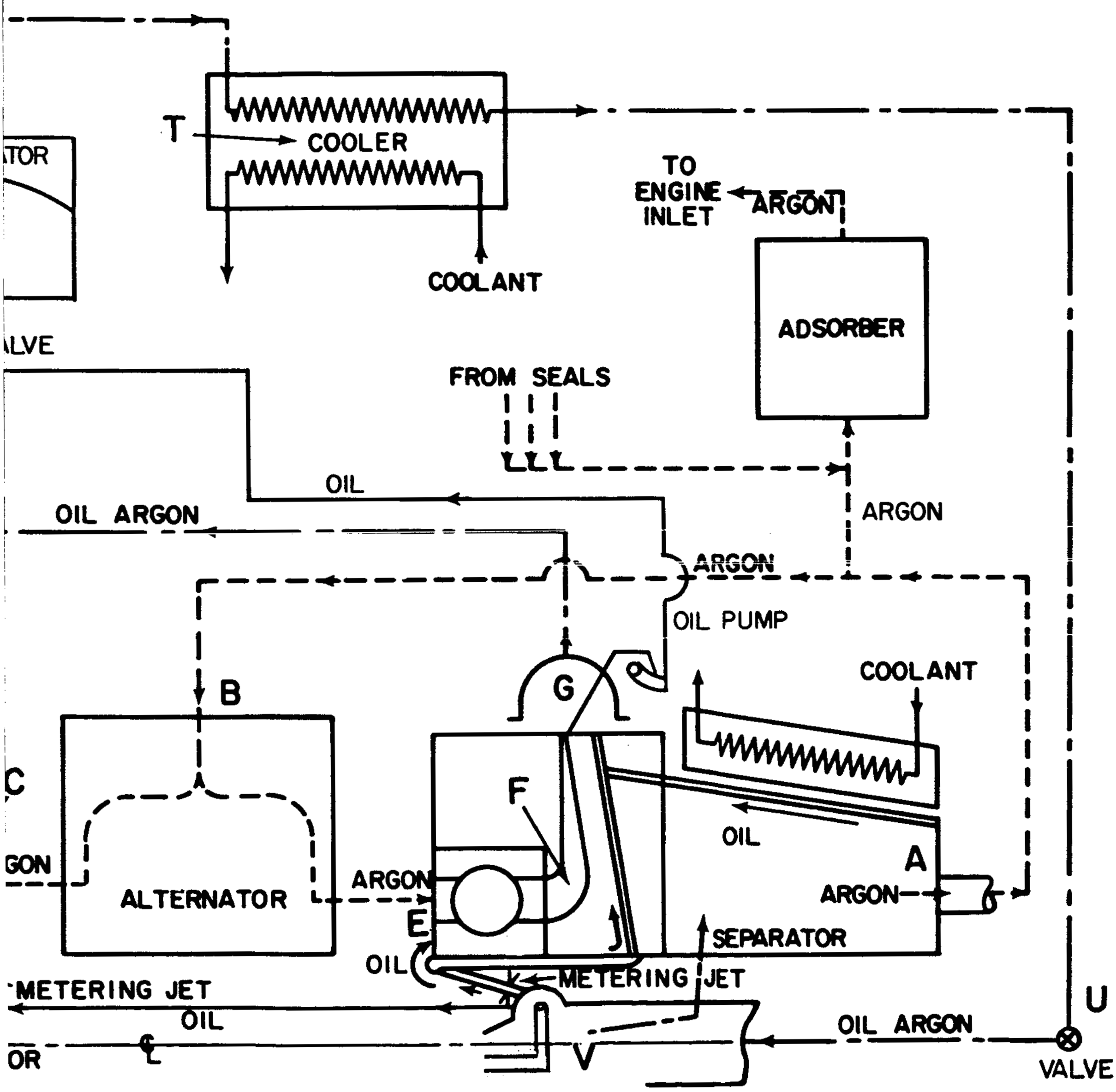


Figure 24 Alternate Brayton-Cycle Rolling-Elem
Oil Scoop-Pump Configuration



ıent Bearing System Schematic

2



OIL

OIL-ARGON (2 PHASE)

ARGON

TABLE 5

Brayton-Cycle Rolling-Element Bearing Lubrication System

Oil Jet Scavenge Pumps in Turbine-Compressor

Summary of Pressure Losses

Horizontal or Zero-Gravity Operation

Station on Figure 3	Description	Oil Flow lb/hr	Argon Flow lb/hr	Press. Loss psi
A-B	line from separator to alternator	-	11.1	0.066
B-C	labyrinth seal	-	5.3	0.186
C-D	roller bearing	-	5.3	0.013
J-K	line from turboalternator to turbine-compressor	10	5.9	0.230
L-M	ball bearing	10	5.9	0.075
N-S	scavenge line	60	6.2	0.194
B-E	labyrinth seal	-	5.8	0.226
E-F	ball bearing	-	5.8	0.009
G-H	line from turboalternator to turbine-compressor	15	6.4	0.150
P-Q	ball bearing	15	6.4	0.080
R-S	scavenge line	80	6.7	0.200
S-T	line to cooler	140	12.9	0.051
T-U	line from cooler	140	12.9	0.126
U-V	scavenge return	140	12.9	0.070
V-A	separator	2	12.9	0.220

TABLE 6

Brayton-Cycle Rolling-Element Bearing Lubrication System

Oil Jet Scavenge Pumps in Turbine-Compressor

Summary of Pressure Losses

Vertical Operation

Station on Figure 3	Description	Oil Flow lb/hr	Argon Flow lb/hr	Press. Loss psi
A-B	line from separator to alternator	-	10.3	0.056
B-C	labyrinth seal	-	4.8	0.152
C-D	roller bearing	-	4.8	0.012
J-K	line from turboalternator to turbine-compressor	10	5.4	0.350
L-M	ball bearing	10	5.4	0.069
N-S	scavenge line	60	5.7	0.142
B-E	labyrinth seal	-	5.5	0.200
E-F	ball bearing	-	5.5	0.009
G-H	line from turboalternator to turbine-compressor	15	6.1	0.280
P-Q	ball bearing	15	6.1	0.076
R-S	scavenge line	80	6.4	0.187
S-T	line to cooler	140	12.1	0.029
T-U	line to cooler	140	12.1	0.104
U-V	scavenge return	140	12.1	0.140
V-A	separator	2	12.1	0.192

The scoops in the turboalternator (Figure 24) impose a drag loss of about 600 watts. A summary of the losses for the lubrication system employing scoop scavenge pumps in the turboalternator is presented in Table 7. The pressure losses in certain sections of the system in Figure 24 are different from the losses in the corresponding sections of Figure 3. These revised performance curves are presented in Figures 25, 26, 27 and 28. The resulting operating conditions with the system in a vertical orientation are summarized in Table 8.

This alternate lubrication system provides greater margin in the uncertainties of two-phase flow, but only at a sacrifice in parasitic losses. Since this alternate system could be incorporated in the turbomachinery with small changes, it will not be pursued further at this time. If the basic system were to be found deficient in experimental evaluation, this alternate could be incorporated by changes in the turboalternator and external plumbing.

A third variation of the lubrication system was examined briefly. This alternate incorporates scavenge pumps similar to those used in SNAP-8[19]. However, the preliminary examination indicated higher parasitic losses and this concept was not considered further.

TABLE 7

Brayton-Cycle Rolling-Element Bearing Lubrication System

Scoop Scavenge System

Power Losses

Type of Loss and Location	Oil Flow lb/hr	Power Loss, watts	
Turbine-Compressor Compartment 1			
bearing heat generation	50	202.1	
seal heat generation	50	200.0	
oil pumping	50	123.0	
argon pumping		10.0	
			535.1
Turbine-Compressor Compartment 2			
bearing heat generation	60	203.5	
seal heat generation	60	200.0	
oil pumping	60	147.5	
argon pumping		10.0	
			561.0
Turboalternator Compartment 1			
bearing heat generation	55	73.0	
seal heat generation	55	55.8	
oil pumping	55	28.8	
argon pumping		2.7	
scoop drag		310.0	
			470.3
Turboalternator Compartment 2			
bearing heat generation	22	108.0	
oil pumping	55	28.8	
argon pumping		2.7	
scoop drag		310.0	
separator drag		7.1	
separator pumping		16.4	
			473.0
Total			2039.4 watts

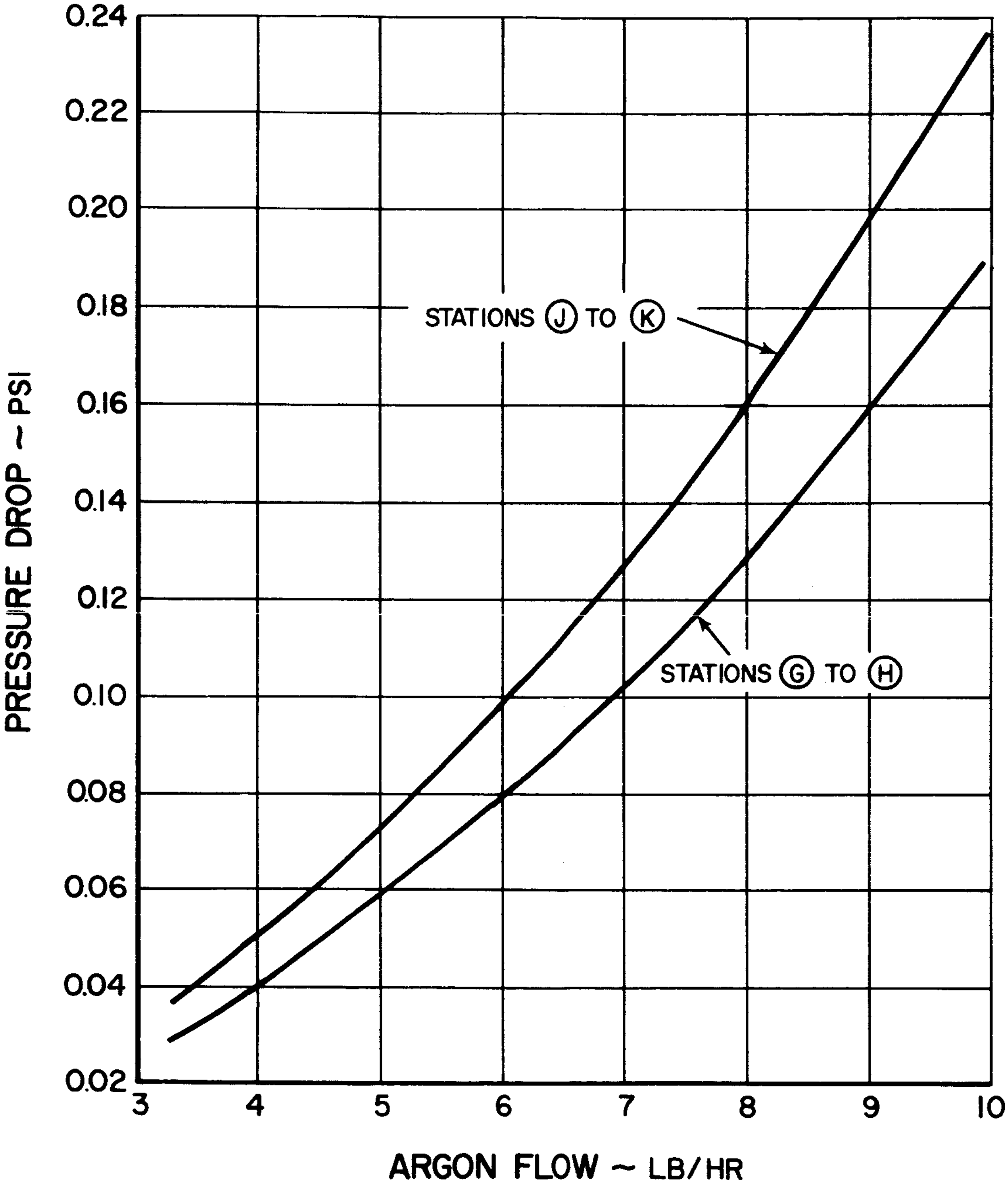


Figure 25 Scoop System. Pressure Drop from Turboalternator to Turbine-Compressor for Argon

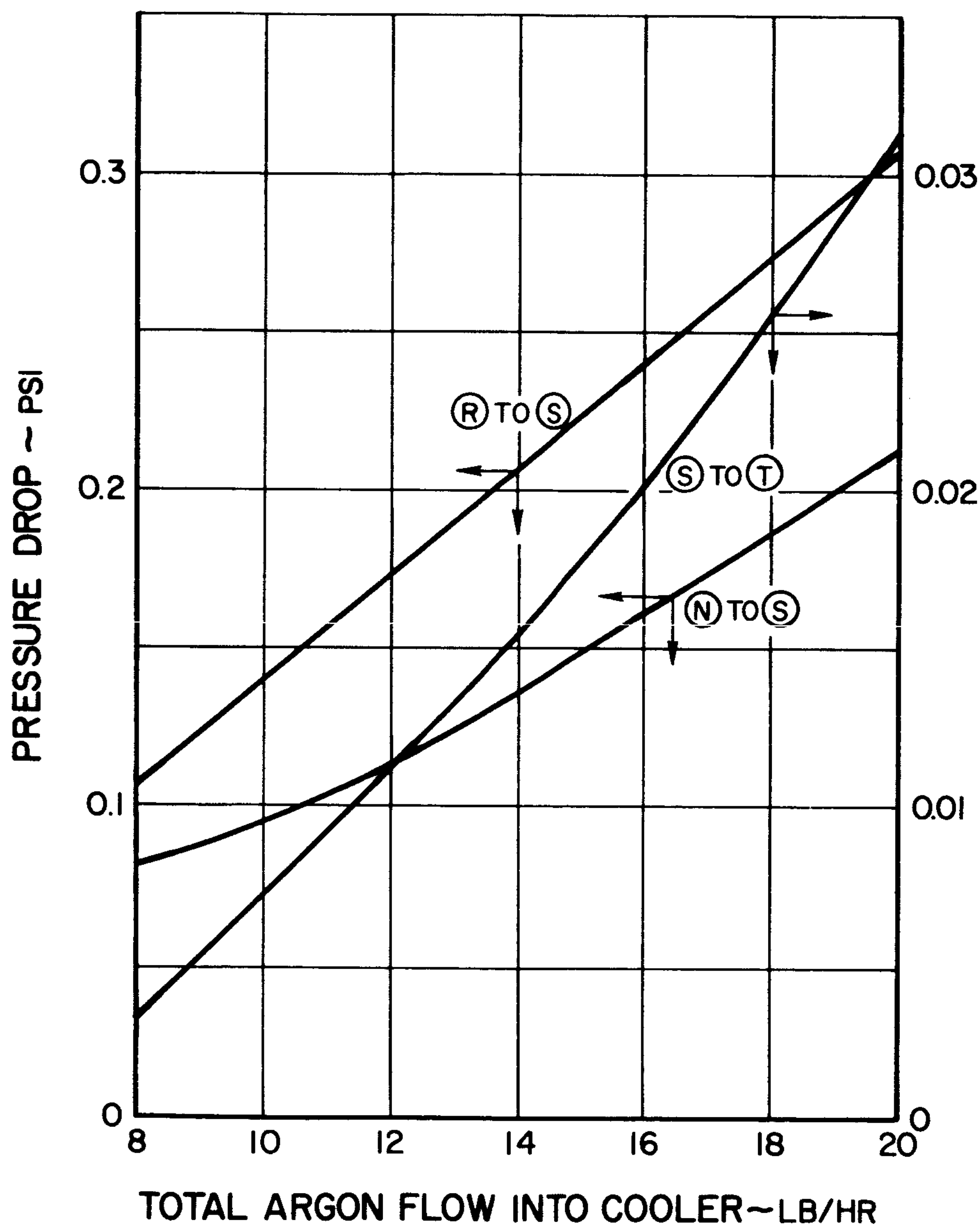


Figure 26 Scoop System. Pressure Drop from Turbine-Compressor to Cooler. Vertical Orientation

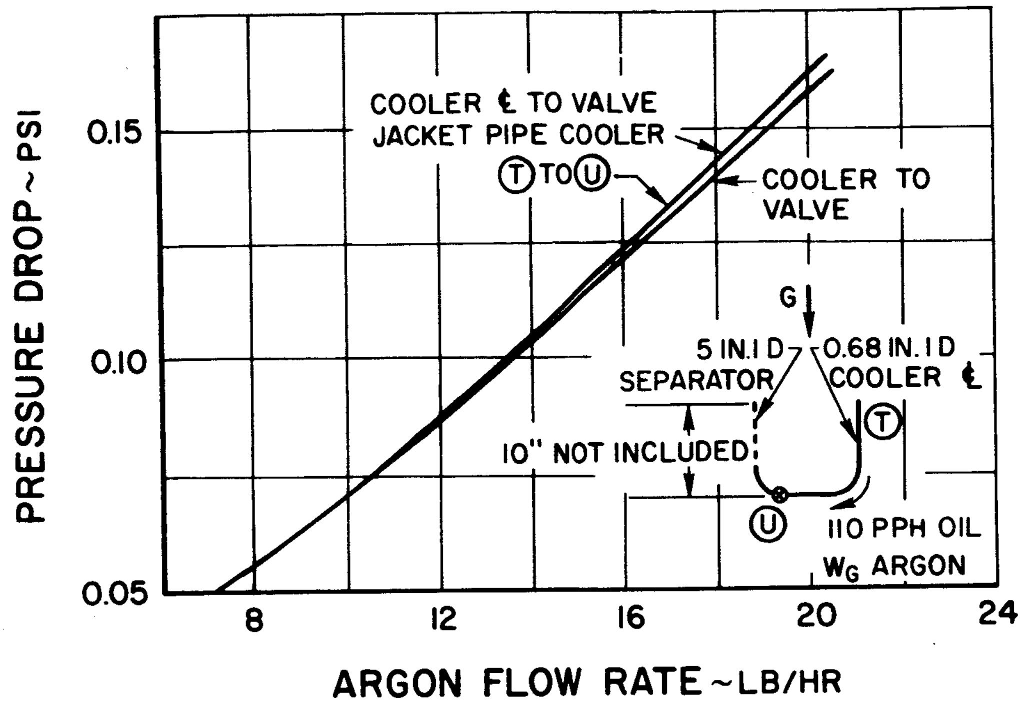


Figure 27 Scoop System. Pressure Drop from Cooler to Separator. Vertical Orientation

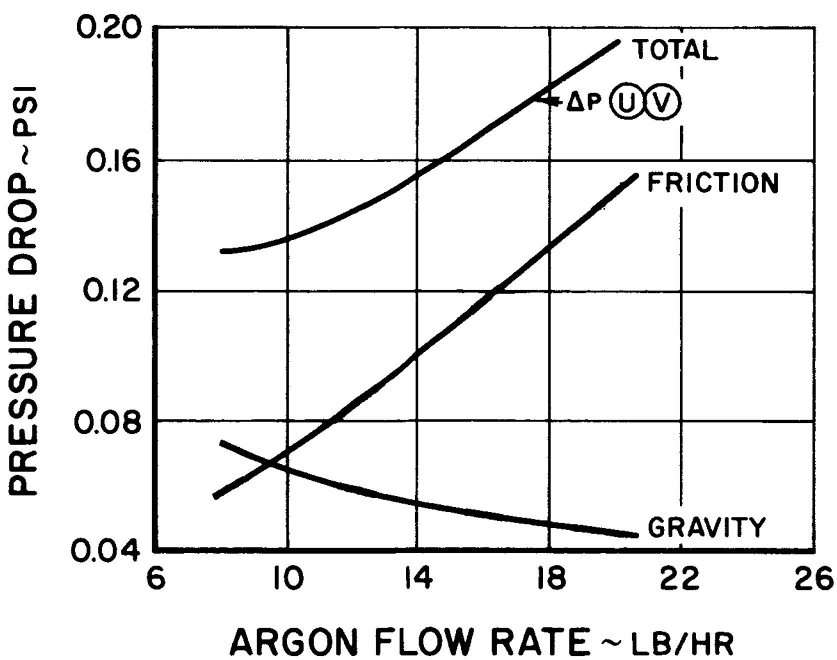


Figure 28 Scoop System. Pressure Drop for Separator Riser

TABLE 8

Brayton-Cycle Rolling-Element Bearing Lubrication System

Scoop Scavenge System

Summary of Pressure Losses

Vertical Operation

Station on Figure 24	Description	Oil Flow, lb/hr	Argon Flow, lb/hr	Pressure Loss, psi
A-B	line from separator to alternator	-	7.5	0.031
B-C	labyrinth seal	-	3.6	0.090
C-D	roller bearing	-	3.6	0.012
J-K	line from turboalternator to turbine-compressor	-	4.2	0.055
L-M	ball bearing	-	4.2	0.048
N-S	scavenge line	50	4.5	0.070
B-E	labyrinth seal	-	3.9	0.104
E-F	ball bearing	-	3.9	0.009
G-H	line from turboalternator to turbine-compressor	-	4.5	0.045
P-Q	ball bearing	-	4.5	0.061
R-S	scavenge line	60	4.8	0.052
S-T	line to cooler	110	9.3	0.006
T-U	line from cooler	110	9.3	0.066
U-V	scavenge return	110	9.3	0.134
V-A	separator	2	0.3	0.109

IV. TURBINE-COMPRESSOR DESIGN

The basic turbine-compressor design task is to adapt a rolling-element bearing system that can be used as an alternate to the gas bearing system which is presently being developed under contract NAS3-4179. The overall design objectives and considerations are: 1) to use a bearing, seal and lubrication system that provides endurance capability with a high degree of reliability for a mission time of 10,000 hours, 2) to use a bearing, seal and lubrication system that has acceptable parasitic losses, and 3) to evolve a mechanical design that utilizes the existing aerodynamics with minimum alterations. In the previous report [1] the general arrangement of the turbine-compressor design was presented. A preliminary critical speed map was determined and the results of a detailed analysis of the turbine-compressor rolling element bearing was presented. During this report period the overall mechanical design was continued. In addition detail effort was concentrated on the rotor dynamics system and the seal designs.

A. Rotor Dynamics

The dynamic motion of the mechanical system which is composed of a rotating assembly, its bearings, seals and bearing support structures was analyzed. The facets of the rotor dynamics considered are the rotor system resonant frequencies (critical speeds), the dynamic shaft displacements due to rotating unbalance, and the resultant bearing loads. In addition, aspects of seal performance and bearing installation and preload are considered relative to the rotor mechanical characteristics.

In the previous quarterly report a preliminary critical speed map was presented. The rotor resonance frequencies were recalculated during this report period to incorporate changes in the design to improve the thermal design and the assembly procedures. The revised critical speed map is shown in Figure 29. Actually, the results are not appreciably different from the earlier results. As in the earlier analysis, for a particular springrate the system passes through the first two criticals, which are essentially rigid-body modes, in accelerating to the design speed of 50,000 rpm.

Further examination of the critical speed map shows that the operating speed of 50,000 rpm is near the second critical mode when the rotor is supported with an effective bearing springrate of approximately 250,000 pounds per inch, corresponding to the turbine-compressor ball bearings. The ball bearing springrate characteristics are shown in Figures 30 and 31 as a function of load and speed. As discussed in Reference 1 the ball bearings require a minimum thrust load of 30 pounds to provide a small skidding margin. To obtain the minimum dynamic response amplitude at the design speed, the spring and damping characteristics of the rotor support system must be selected to provide critical speeds which are well removed from the operating speed. The problem, then, of designing a rotor support is to select a method which will provide the required bearing support springrate and damping as well as provide axial movement to accomplish bearing thrust loading.

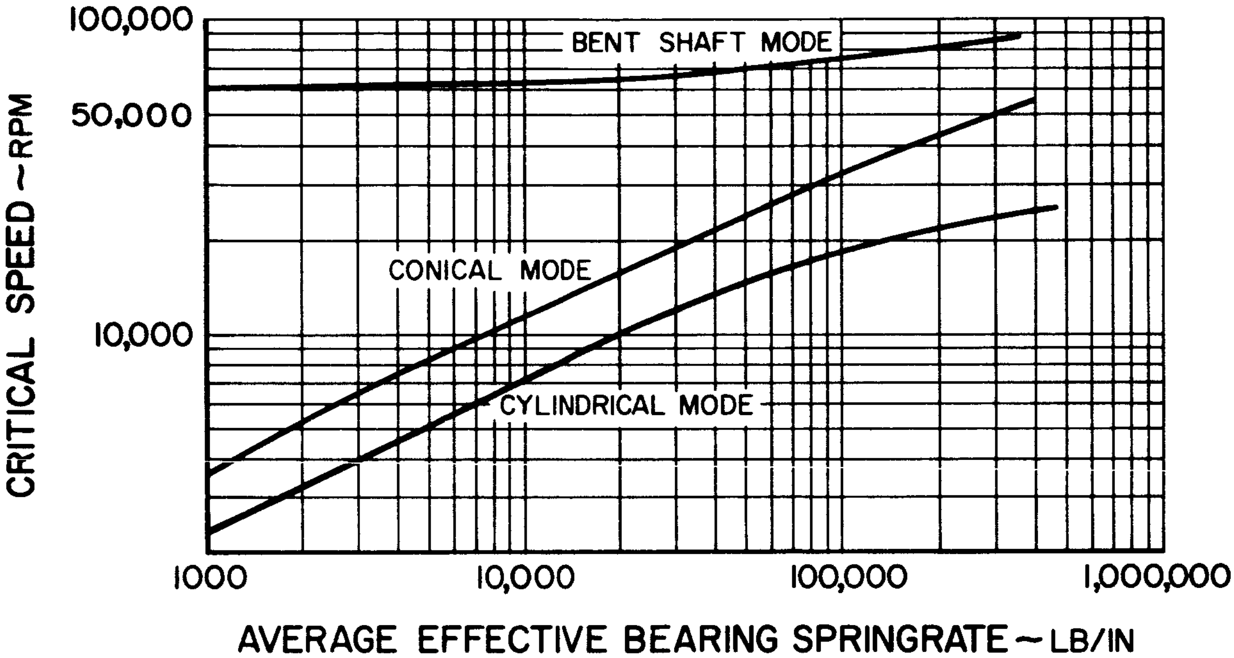


Figure 29 Critical Speed Map for Turbine-Compressor

Analysis of possible methods of providing the required rotor support springrate, damping and bearing thrust load led to the selection of an oil squeeze film type of bearing support. The configuration of this support is shown in Figure 32. The outer race of the bearing is mounted in a nonrotating cylindrical member. The oil support film is contained in the controlled clearance between the outside diameter of the cylinder and the bore of the compressor case. Radial forces from the rotor are transmitted through the ball bearings and support cylinder to the squeeze film. The oil film provides a nonlinear support as well as damping. Coil springs located at the end of the cylindrical

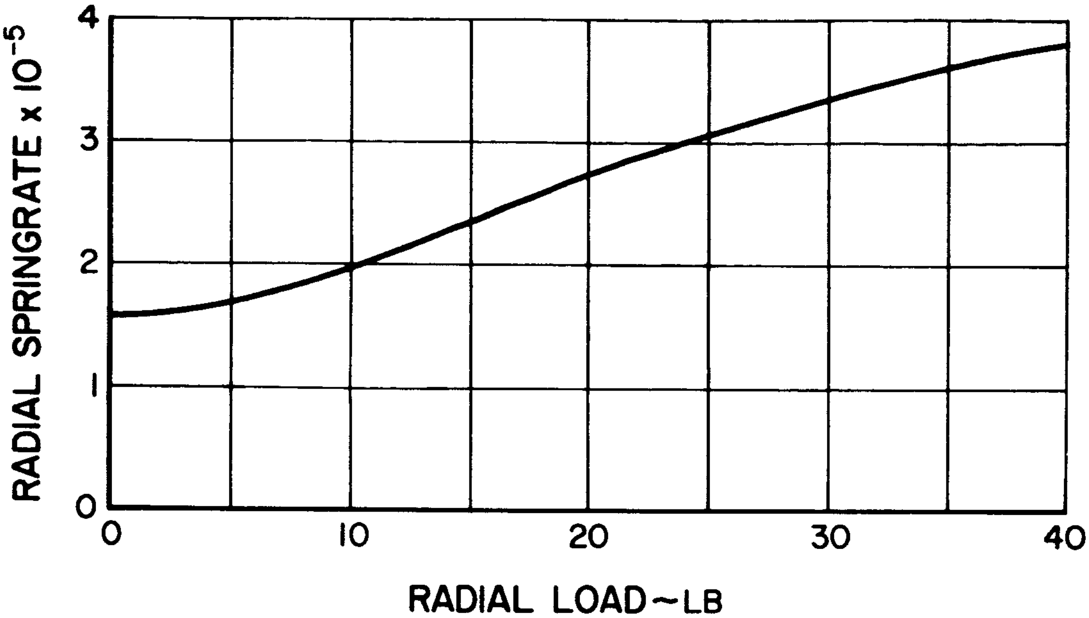


Figure 30 Turbine-Compressor Bearing Radial Springrate vs Radial Load at 50,000 rpm and Thrust Load of 30 lbs

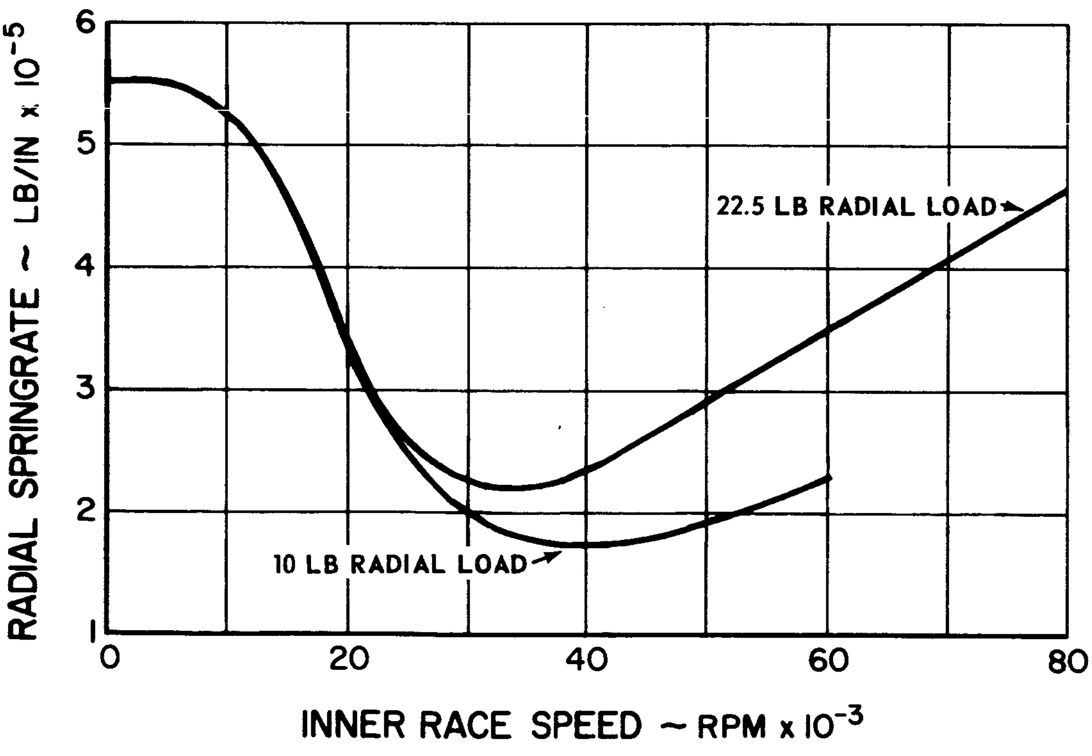


Figure 31 Turbine-Compressor Bearing Radial Springrate (Thrust Load 30 lbs) vs Inner Race Speed

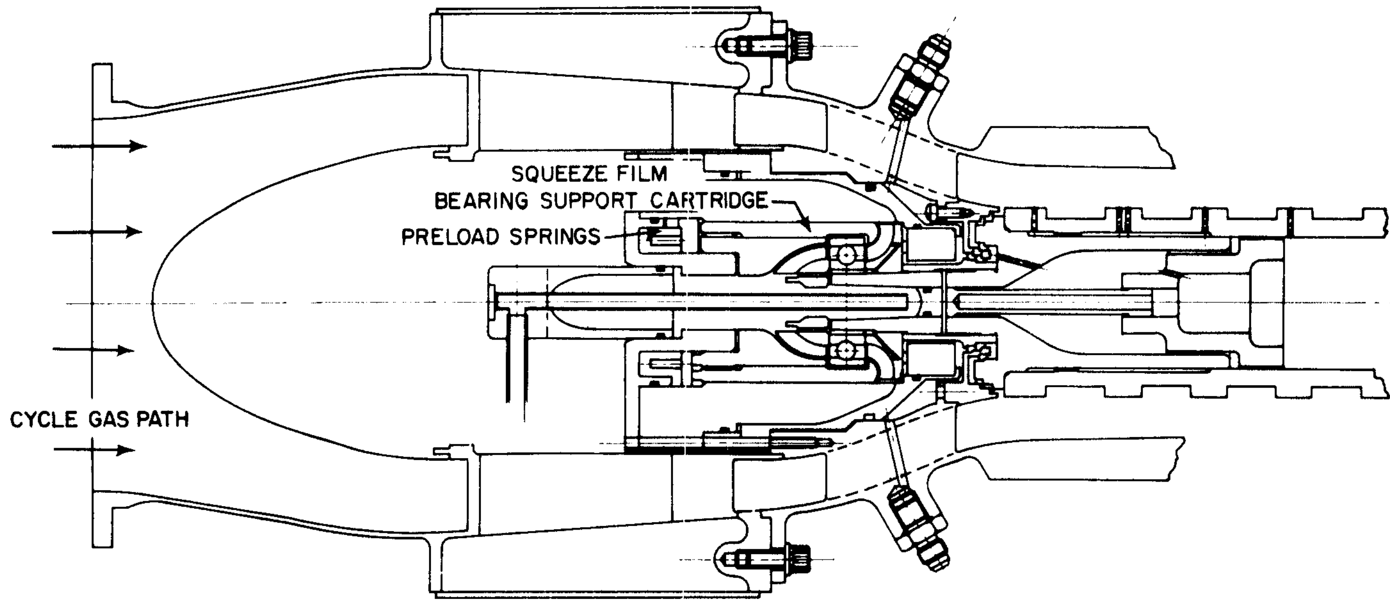


Figure 32 Turbine-Compressor No. 1 Bearing Section

member in which the bearing is mounted provide the necessary bearing pre-load. This feature also provides for axial movement of parts due to thermal expansion. The combination of the squeeze film and the spring loading into one assembly provides a relatively simple design.

The dynamic rotor response characteristics at the turbine-compressor rear bearing, the most critical location, are shown in Figure 33. The total deflection is due to the deflection in the ball bearing and the oil squeeze film. The overall springrate of the rotor support consists of the springrate of the ball bearing and the oil squeeze film which operate in series. As shown on the curve, the total movement of the rotor at the rear support is less than 1 mil at 50,000 rpm with 0.002 ounce-inch unbalance which is the maximum amount of unbalance anticipated at the end of 10,000 hours of operation. The effectiveness of the oil squeeze film in reducing the overall rotor support springrate to obtain critical speed margins is shown by Figure 34. This curve of the relationship of rotor speed to critical speed indicates that the oil film support provides a system having a safe margin between the operating speed of 50,000 rpm and the second and third critical modes.

As can be seen from the results of Figure 34, and by referring back to the critical speed map of Figure 29, the effective rotor support springrate with the squeeze film is near 100,000 pounds per inch instead of near 250,000 pounds per inch, the rate with the bearings alone.

A discussion was presented in the previous report [1] relative to the design features of the bearing cavity at the turbine end of the turbine-compressor. The bearing location was shifted outward along the shaft and the envelope increased in size radially to provide a cooler and larger bearing cavity. This necessitated a slight refairing of the turbine exhaust duct. Performance changes resulting from the refairing are considered to be negligible. A sectional view of the modified bearing cavity including the squeeze film bearing support is shown in Figure 35.

In order to reduce the heat flow from the ductwork surrounding the cavity it was necessary to install a thermal barrier. As indicated in the lubrication system of this report, more oil is circulated through this cavity than through the front cavity to aid in controlling the temperature of bearing and seal. The expected temperature pattern of this region is shown in Figure 36. Examination of this figure shows that the temperature of the bearing inner race is 371°F and that of the outer race is 391°F. Also the temperature of the seal is estimated to be 430°F. Additional cooling and insulation could be added if necessary.

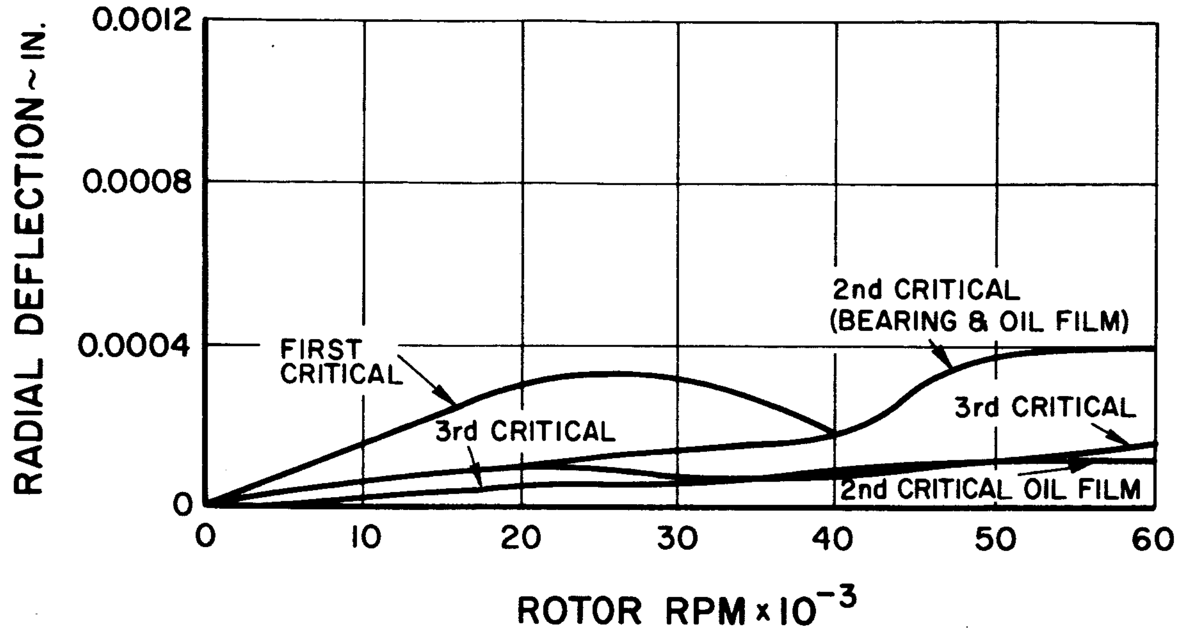


Figure 33 Dynamic Rotor Response at Turbine-Compressor Rear Bearing

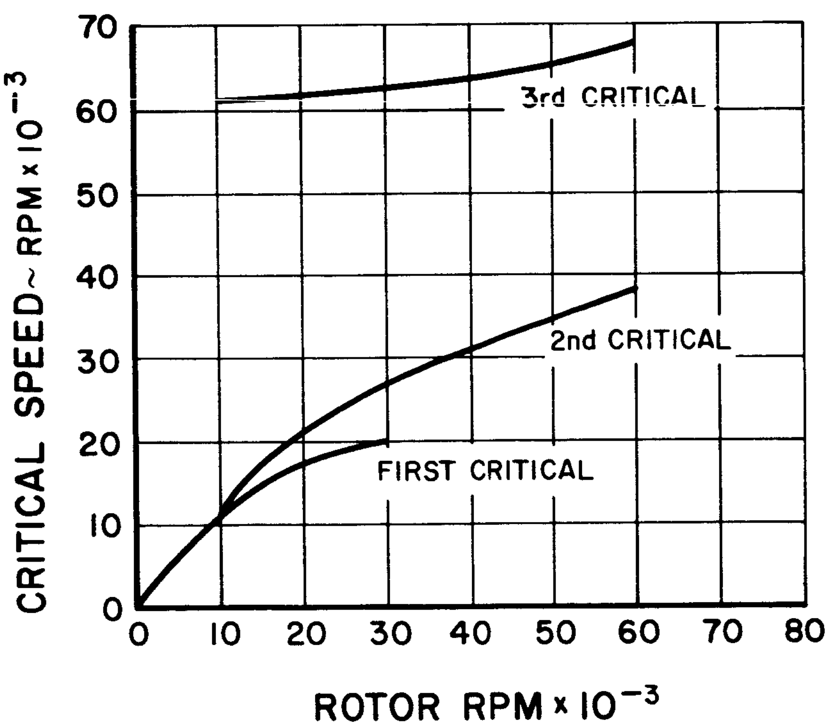


Figure 34 Relationship between Rotor Speed and Critical Speed for Turbine-Compressor with Squeeze Film Support

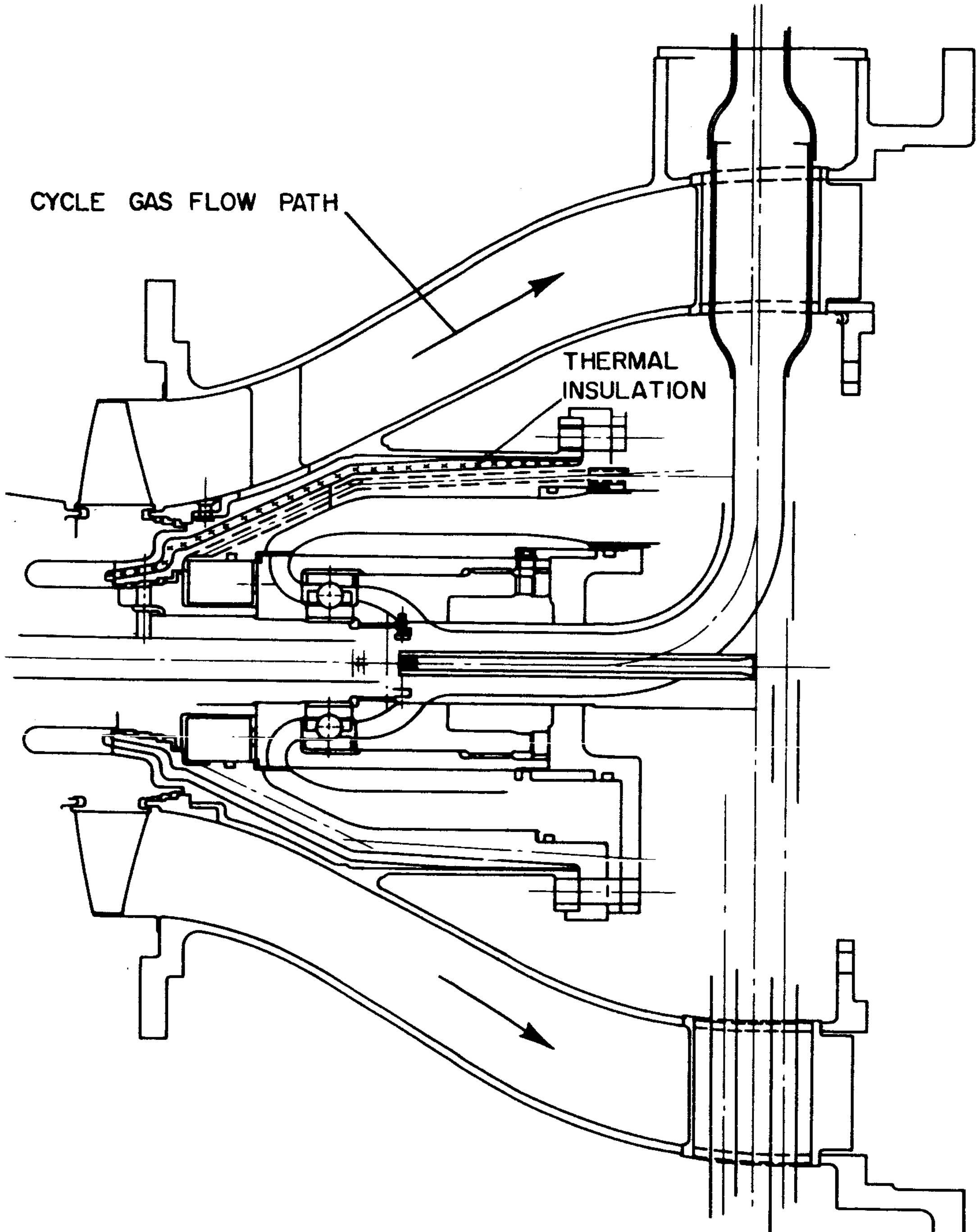


Figure 35 Turbine-Compressor No. 2 Bearing Section

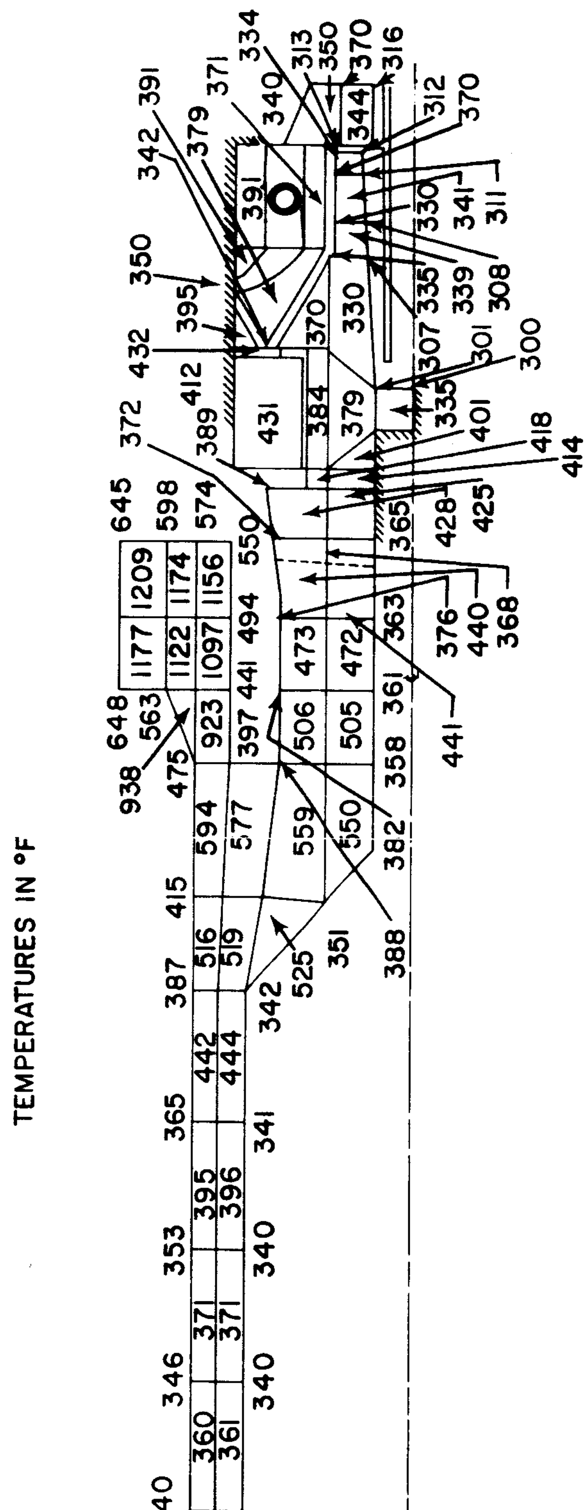


Figure 36 Typical Temperature Pattern in Turbine-Compressor No. 2 Bearing Region

Installation of the squeeze film bearing support is similar to that at the front bearing location. The major difference is that the thrust load for this bearing is obtained from the springs located in the front bearing position. The inner race is attached to the shaft in compression with a nut. Thrust is taken through the inner race and through the balls to the outer race. The outer race is inserted in the squeeze film cylinder which is supported in the compressor case.

Oil is introduced to the bearing via a small pressure tube located concentric to the rotor. As discussed above (see lubrication system schematics), the oil is pumped radially outward underneath the bearing race and then out through the seal plate. Gas is fed into the cavity in a tube surrounding the oil supply tube. This oil-laden gas is pumped through the bearing into the seal plate. The cooling oil flow and oil-laden gas join in the seal plate impeller and are then discharged into the scavenge line.

B. Seals

The basic sealing concept being developed for all main shaft bearing compartments consists of two labyrinth seals adjacent to a face seal. High-pressure argon is fed between the labyrinth seals to provide a pressure differential across the face seal and to provide a purge for oil that might leak past the face seal. This pressure differential aids in containing the oil within the lubrication system. This feature also provides a buffer zone between the cycle gas and the bearing and seal cavities to reduce cycle gas contamination. A system is provided to recover, clean up, and re-use the bleed gas as well as the gas which leaks past the seals. A further requirement of these designs is that there be no oil leakage from the bearing compartment at shutdown.

The primary consideration in designing the face seals for this application is to weigh the performance characteristics of several possible designs against the complexity and degree of risk involved. Seal configurations being considered will minimize oil leakage, gas leakage, and power consumption consistent with the long term durability and reliability requirements of the Brayton-cycle power system.

Four face seal designs are being considered in detail for possible evaluation and use in the turbine-compressor. All four designs have the same overall envelope to permit interchangeability during evaluation and eventual use. The designs can be categorized as 1) dry face, 2) oil-lubricated, 3) gas-lubricated, and 4) controlled-clearance. Each design becomes a positive contact seal at shutdown. The first design is based upon the accepted general practice of having a carbon ring bear against a rotating metal seal plate. In this design, the carbon grade, carbon geometry, and face loading are the primary variables. Compatibility of the rotating metal seal plate is also considered. The stringent low leakage requirements indicate that the best choice for the secondary seal is a damped metallic bellows. A bellows is a zero-leakage seal which accommodates a misalignment and axial movement due to thermal gradients, mechanical fitup and wear. The bellows proposed for this design, as well as for the other designs, incorporates a damper which improves the reliability and life of the bellows. The manner in which this type of seal would be incorporated into the turbine-compressor is shown in Figure 35. A predicted thermal map for this type of seal in the turbine-compressor is shown in Figure 36 for the Number 2 location. As shown in the thermal map, the maximum temperature expected is in the order of 400 to 430°F. It is estimated that this seal would consume about 200 watts and would leak about 0.3 pound per hour of argon at the design speed of 50,000 rpm. A wear allowance of 0.030 inch is provided.

The second seal design being considered for the turbine-compressor is a modification to the dry-face design. This second type, called a wet-face design, incorporates oil passages through the rotating metallic seal plate which directs oil to the contact area between carbon and the metallic seal plate. The oil introduced forms a film between the mating surfaces. The net result is to decrease the coefficient of rubbing friction and increase the hydrodynamic drag. Other features of this design, such as the bellows, are essentially the same as the first design discussed above. Relative to the dry-face design, gas leakage will be lower, and wear will be greatly reduced. Power consumption will be of the same order of magnitude as that of the dry-face design and the thermal maps are essentially unchanged.

The third design is of the hydrodynamic lift variety. The contact face is modified to provide hydrodynamic lift capability through a spiral-groove configuration based upon the Whipple gas bearing design. The spiral-groove outward-pumping design is favored because of its inherent high film stiffness. The seal face loading of the selected groove configuration as a function of film thickness is shown in Figure 37. The film stiffness characteristics are shown in Figure 38. Film thicknesses of 0.0003 to 0.0004 inch are needed to assure an adequate margin between resonant frequency and operating frequency. The outward-pumping feature will oppose oil weepage and reduce operating temperature. Power consumption is expected to be less than 50 watts. Argon leakage flow of 0.2 to 0.3 pound per hour is anticipated and wear is expected to be very low.

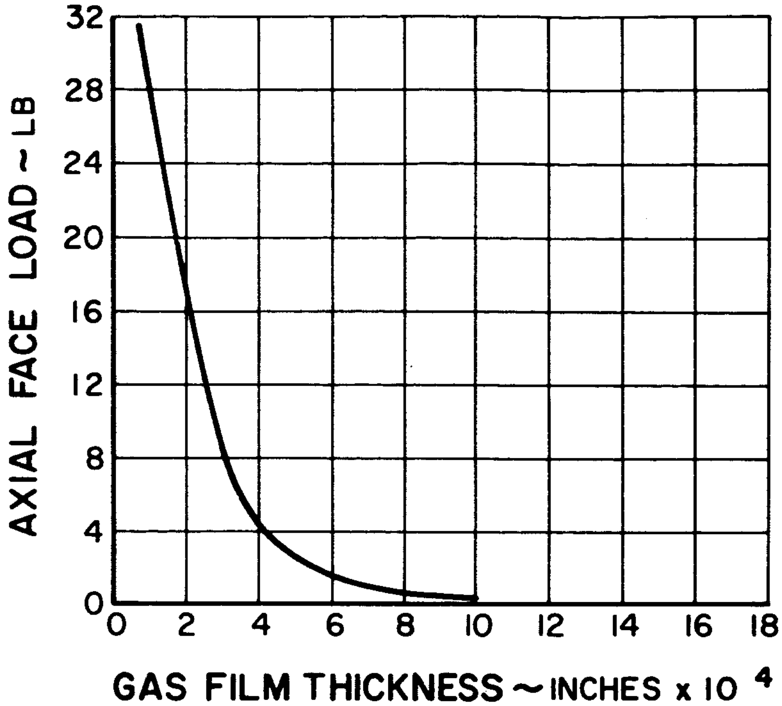


Figure 37 Turbine-Compressor Hydrodynamic Gas Film Seal. Spiral Groove Configuration. Load-Carrying Characteristics

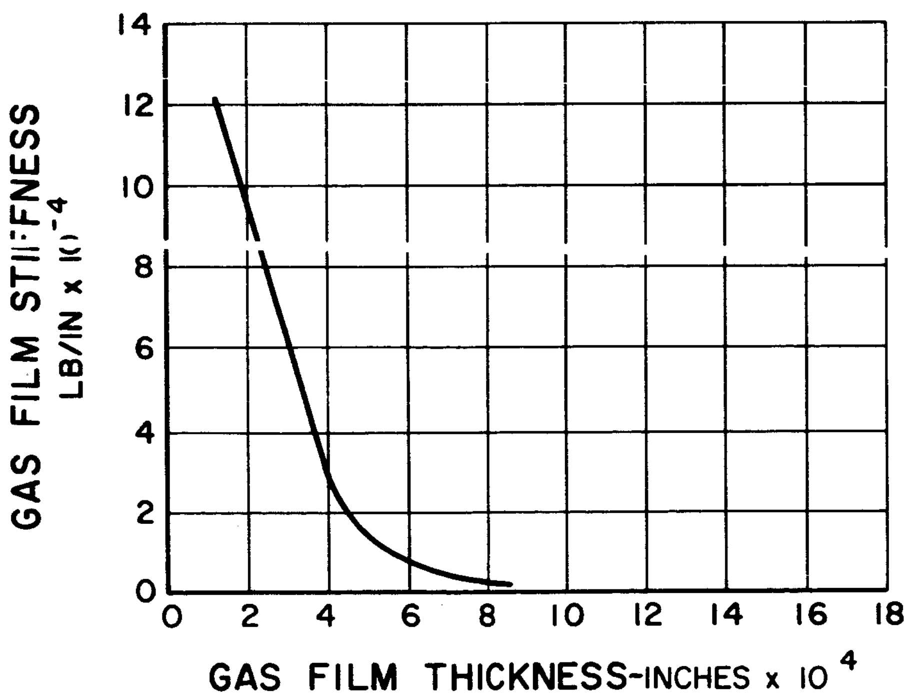


Figure 38 Turbine-Compressor Hydrodynamic Gas Film Seal. Spiral Groove Configuration. Gas Film Stiffness Characteristics

The fourth design being considered for the turbine-compressor is a floating labyrinth bellows seal. The labyrinth is a floating carbon ring which has a close radial clearance with the rotating shaft. The carbon ring is free to move axially along the shaft and seat on the shaft-rotating seal plate on one side and on a retractable bellows-supported seal plate on the other. Under static conditions, this seal resembles a positive contact face seal. That is, the bellows acts as a spring and clamps the carbon ring between two seal plates, one attached to the shaft and the other to the bellows. When operating under pressure and temperature, the nonrotating seal plate retracts, followed by the carbon ring, allowing a small clearance space between the shaft and the carbon ring to control the leakage. The expansion rate of the carbon ring is controlled by a metal shroud on the carbon outer diameter and thereby matched to the shaft expansion. This design has a potential of low leakage (0. 2 - 0. 3 pound per hour) and low power consumption (less than 50 watts) with good reliability. Oil leakage characteristics, however, are not well known.

The test phase of the seal program calls for operation of three turbine-compressor seal designs. These tests will verify the design principles and establish actual leakage rates and power consumption characteristics. It may be possible to reduce the overall system power consumption several hundred watts with either the gas-lubricated or controlled-clearance type seal. However, experience with these seals is limited and their oil leakage characteristics are not well known. Therefore, the estimated seal power consumption used in the system analysis is based on the more conventional dry and wet-face seals.

V. TURBOALTERNATOR DESIGN

The basic turboalternator design task under this program is to provide a rolling-element bearing system design that can be used as an alternate to the gas bearings presently being developed under Contract NAS3-6013. The overall objectives of the rolling-element bearing design for the turboalternator are identical to those of the corresponding turbine-compressor design task discussed in Section IV.

The general arrangement of the design is shown in Figure 39, which is essentially the same as the concept presented in Reference 1. The unit consists of three main sections, the alternator rotor, the drive turbine, and the oil-gas separator. The alternator is straddle-mounted between a ball and a roller bearing. The turbine is overhung from one end of the shaft and the separator is overhung from the other end.

During this report period refinements were made in the rotor design, mainly in the overhung oil-gas separator. The oil-gas separator shown in Figure 40 consists of a cylindrical housing assembly attached to the end of the alternator shaft adjacent to the main shaft ball bearing. This cylindrical housing contains a porous metallic matrix which provides a surface for the oil to coalesce on as the oil-gas mixture is centrifuged. An oil scoop is located at the center of the unit to pump oil from the annulus in the shaft. The small amount of oil not separated by centrifugal effects and the scoop is separated in the matrix and pumped along the tapered periphery of the matrix cylinder to the main slinger. The slinger adjacent to the ball bearing pumps this oil as well as the oil from the ball bearing out of the turboalternator. Figure 40 shows that stationary cooling passages are incorporated around the circumference of the separator housing. The purpose of this coolant is to reduce the temperature of the oil-gas mixture as it passes through the separator. The cooling removes oil from the gas as well as cooling the gas before it passes on to the adsorber and turboalternator rotor purge lines.

Design details and refinements were incorporated in the main shaft support in the bearing and seal areas. As shown in Figure 1, only one seal is required, that at the turbine end of the rotor. No seal is required at the other end since the bearing cavity is connected to the separator. A face type contact dry carbon seal was selected to run against an oil-cooled seal plate. Heat generation for the seal is estimated to be 56 watts. The arrangement of the seal envelope can be modified to accept a wet-face seal at a later date, if desired. The

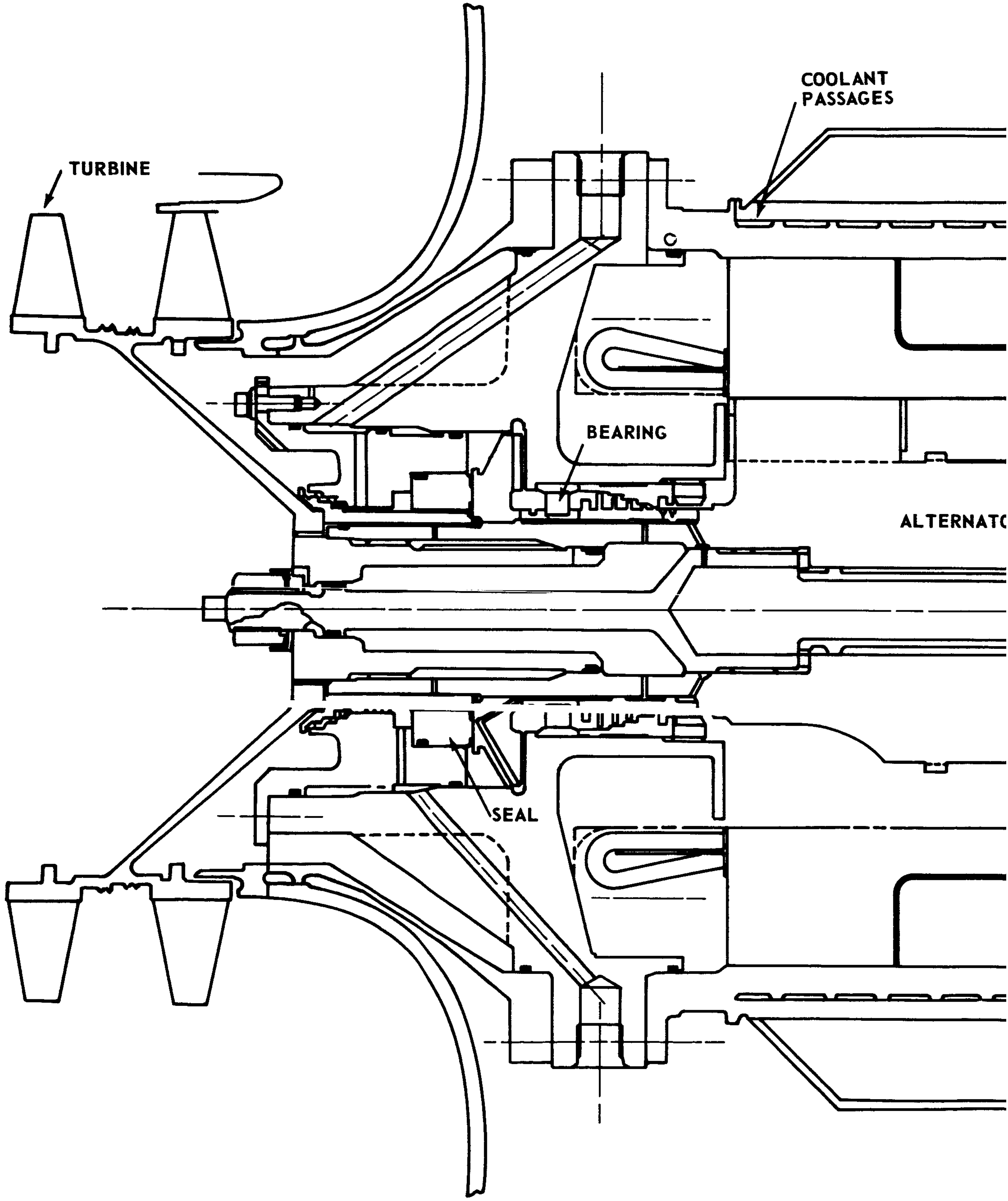
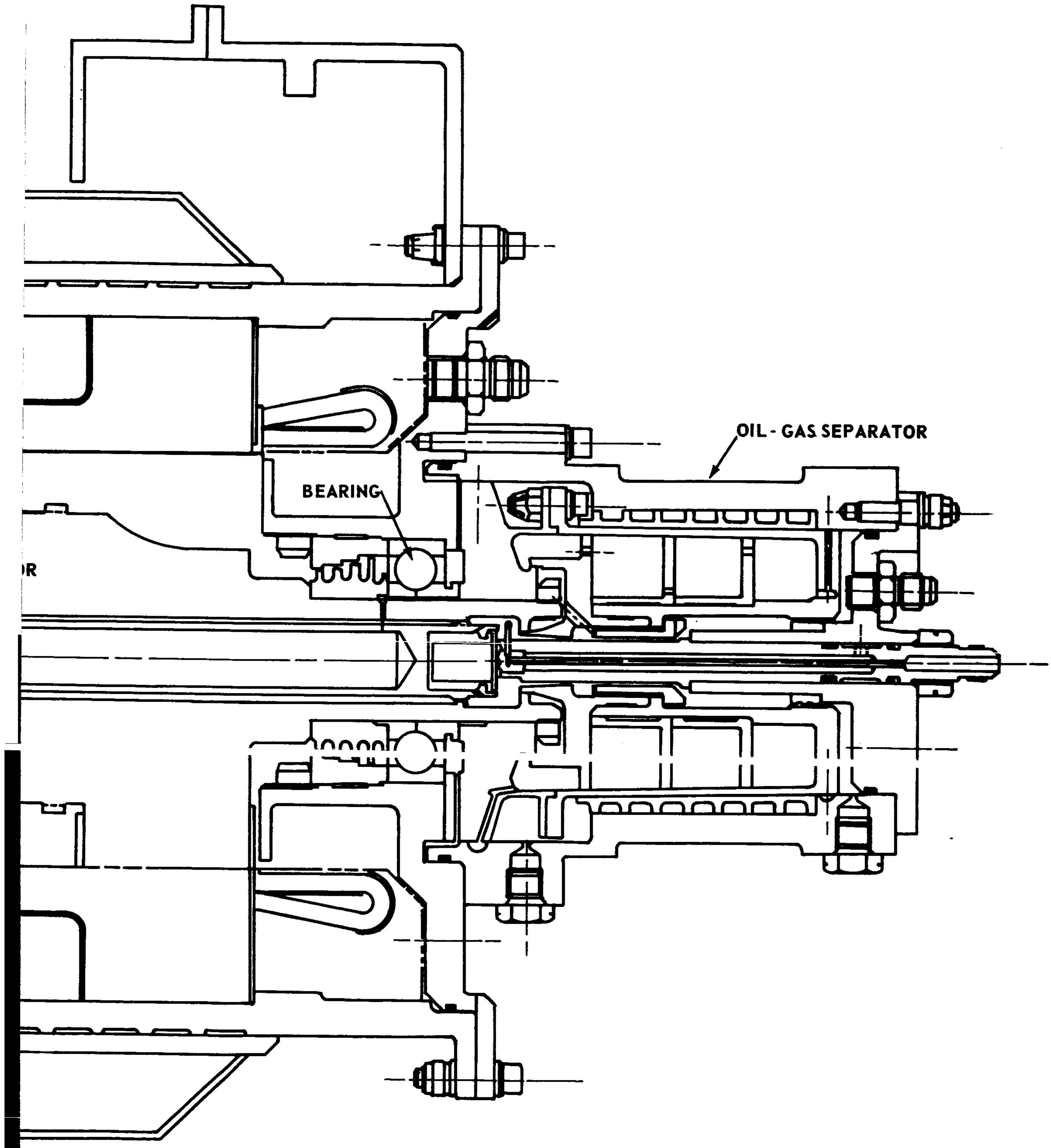


Figure 39 Turboalternator Longitudinal Section


OIL - GAS SEPARATOR
BEARING
OR

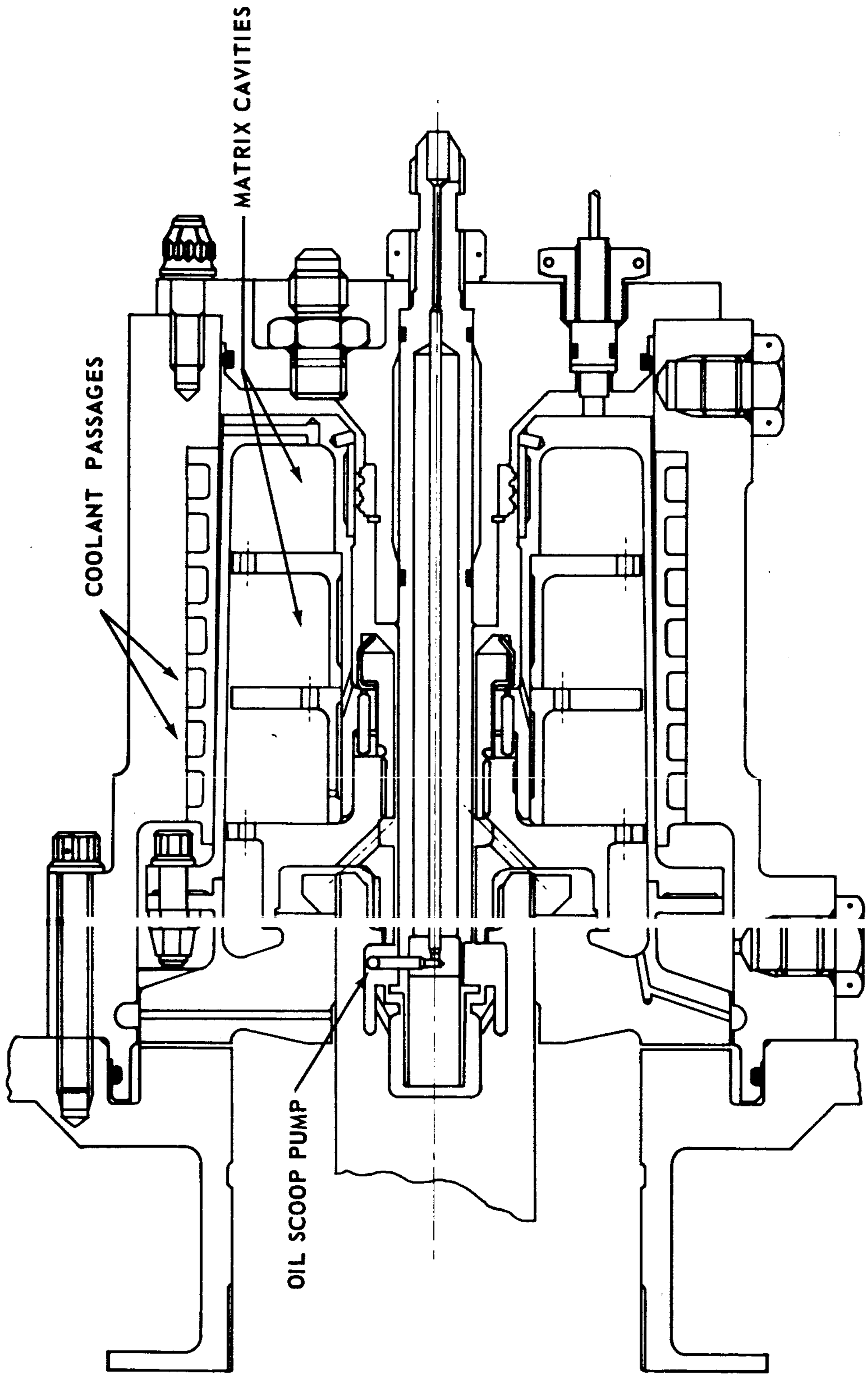


Figure 40 Turboalternator Oil-Gas Separator

thermal environments predicted for the bearing and seal regions at each end of the rotor are shown in Figures 41 and 42. Figure 41 shows the temperature pattern for the roller bearing and seal at the turbine end of the rotor. All temperatures are well within reasonable limits for the selected materials and anticipated operating conditions. The temperature pattern shown in Figure 42 for the separator end of the turboalternator also indicates that all temperatures are well within established limits. The anticipated cooling pattern for the oil-gas mixture as it passes through the separator is shown in this figure. The temperature of effluent gas from the separator is estimated to be reduced to approximately 100°F.

In addition to the design work discussed above, details related to machining, balancing and assembly were worked out. The rotor is designed so that all close tolerances and concentricities can be held by minimizing the number of machine setups and machining operations. Balancing of detail parts as well as subassemblies is required during assembly. As a final step the complete rotor assembly is balanced and then checked at the design speed of 12,000 rpm.

TEMPERATURES IN °F

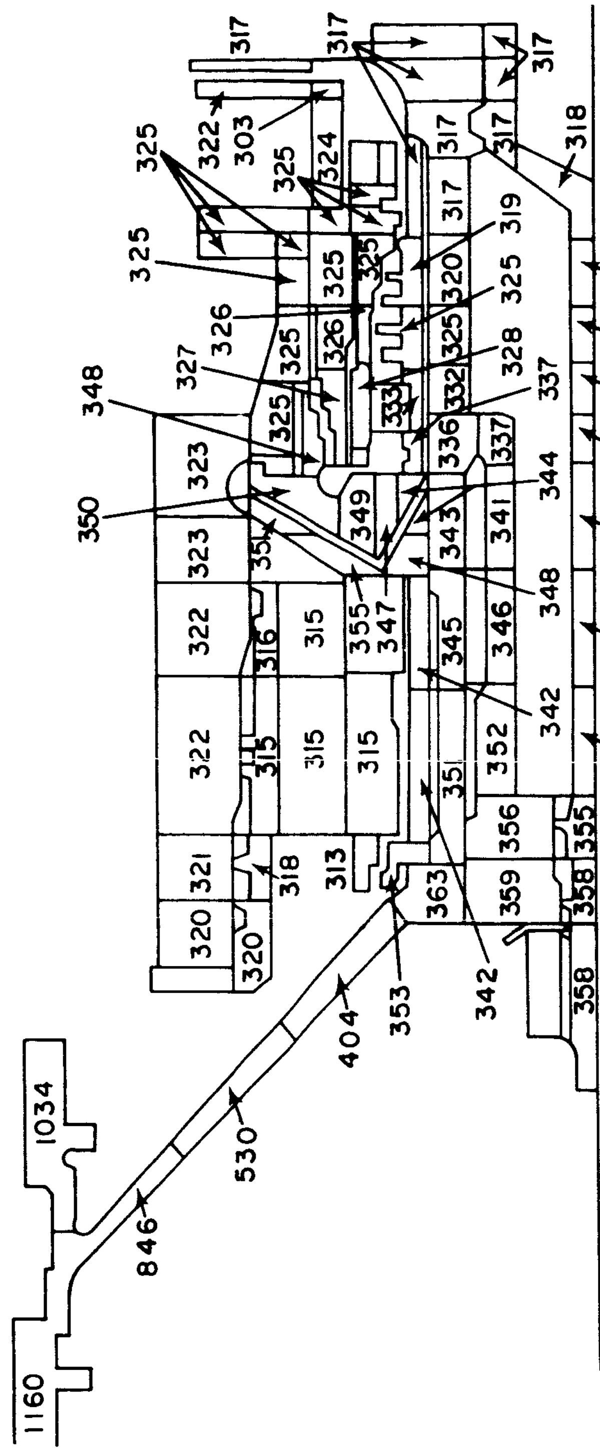


Figure 41 Thermal Map of Turbcalternator No. 1 Bearing Region

TEMPERATURES IN °F

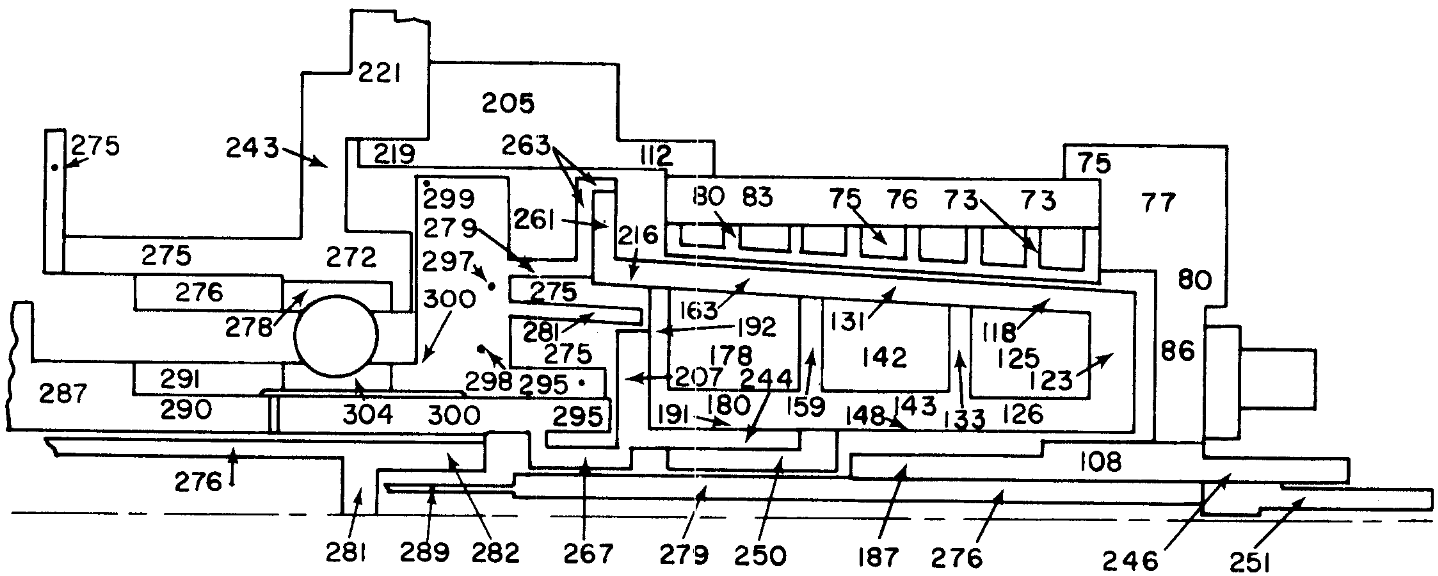


Figure 42 Thermal Map of Turboalternator No. 2 Bearing Region

VI. BEARING-SEAL-SCAVENGE RIG

Performance investigation of selected elements of the oil-lubricated rolling-element bearing system will be conducted as a part of the overall program. Turbine-compressor components were selected for experimental evaluation because at the higher speed and in the more confined space, they were considered to be more critical than similar elements in the turboalternator. In order to conduct the experimental investigations, test rigs are required. Since the turbine-compressor bearings, seals and scavenge pumps are similar in size and are located adjacent to each other, one basic type of test rig was selected to test each element separately or in combination with a minimum number of changes. The arrangement for seal test evaluation is shown in Figure 43 and that for bearing and scavenge test in Figure 44. In both configurations the drive turbine and interconnecting housings are common, only the test section is different.

In the seal test configuration the major considerations were 1) to provide an environment for the seal that duplicates the turbine-compressor environment, and 2) to provide a method of evaluating seal performance, leakage in particular, with a high degree of accuracy. The rig is made up of three assemblies, an air drive turbine, a test adapter, and the test region. The drive turbine is designed to operate on shop air to provide approximately 5 horsepower at speeds up to 60,000 rpm. The function of the test adapter is merely to provide a structural support between the drive turbine and the test region and to provide a means of controlling pressures in the test region which simulate those in the turbine-compressor. The purpose of the test region is, as the name implies, to house the test seal. The rotor in this version of the rig has a first critical speed of 67,000 rpm.

Performance tests of three turbine-compressor seal designs will be conducted. In each test argon leakage, oil leakage, oil flow rate, oil inlet temperature, oil outlet temperature, oil supply pressure, and gas pressure on each side of the seal will be measured. Seal power consumption will be estimated from test data. Tests will be run at various shaft speeds, including 12,000 and 50,000 rpm. Gas leakage will be measured under static conditions. Seal and seal plate wear and distortion will be measured. Photographic observations will be made as appropriate to record wear pattern, oil sludging and general appearance. The seal test program will be concluded by testing the most promising design for 100 hours.

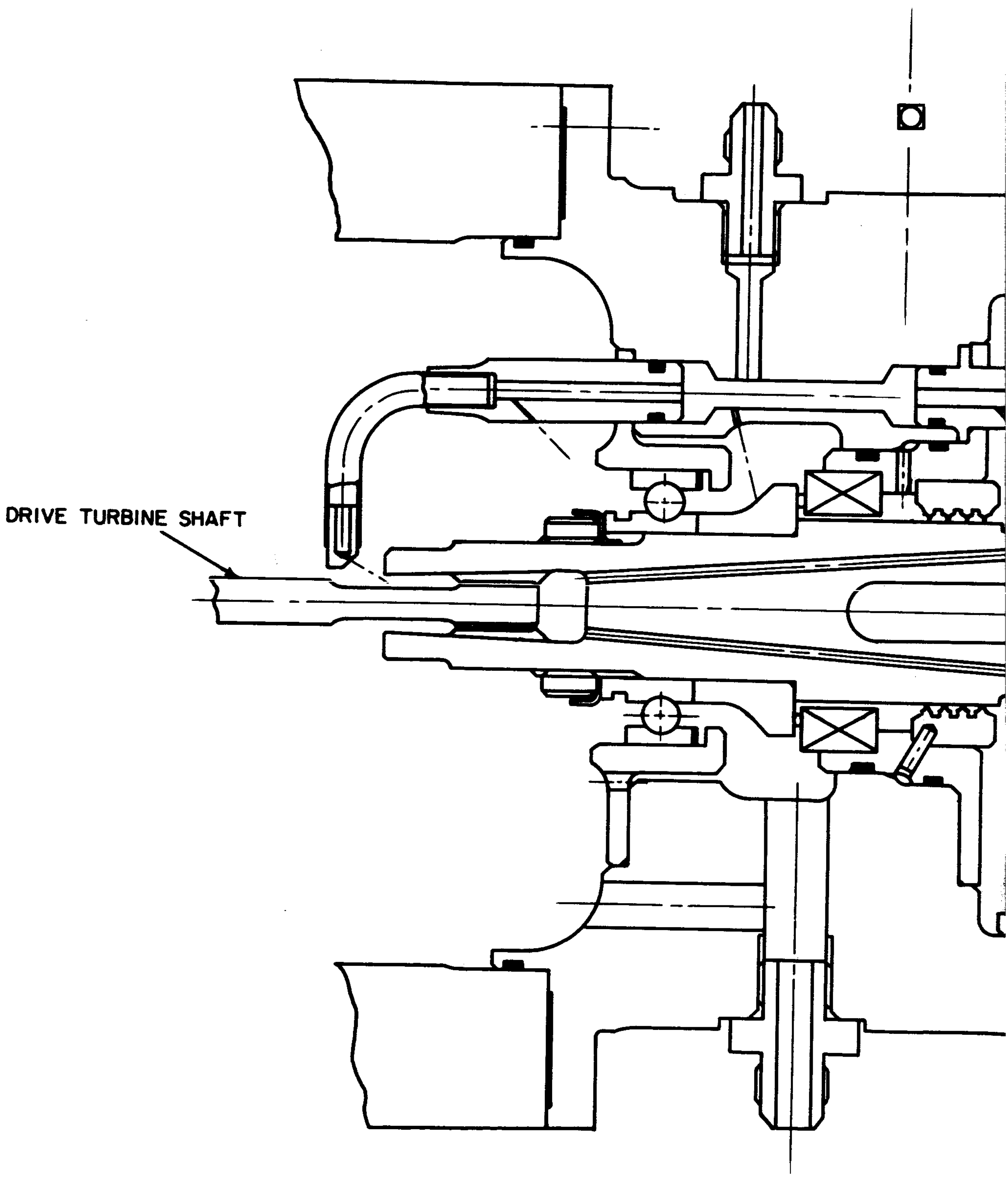
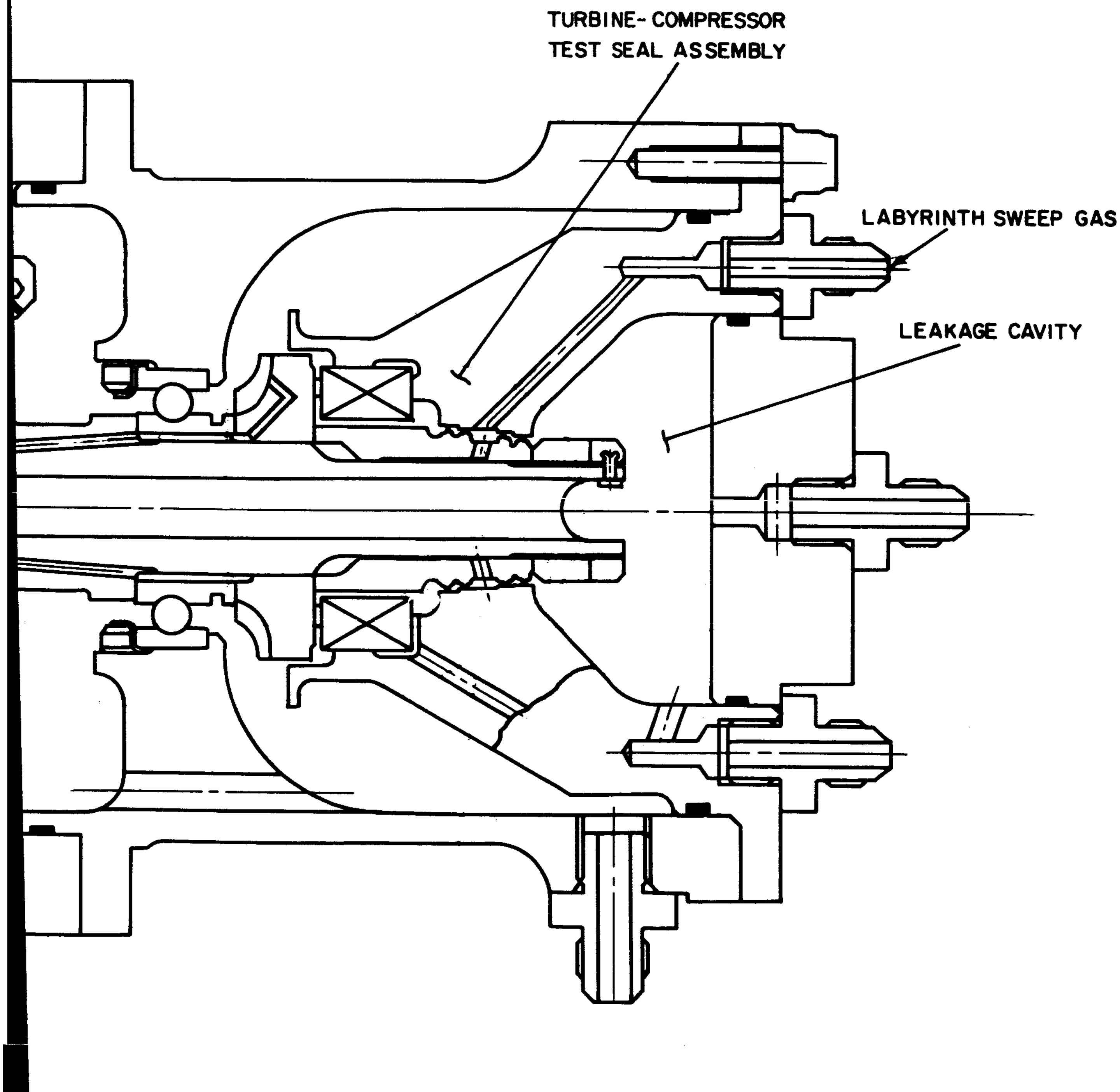


Figure 43 Turbine-Compressor Seal Test Rig


TURBINE-COMPRESSOR
TEST SEAL ASSEMBLY
LABYRINTH SWEEP GAS
LEAKAGE CAVITY

2

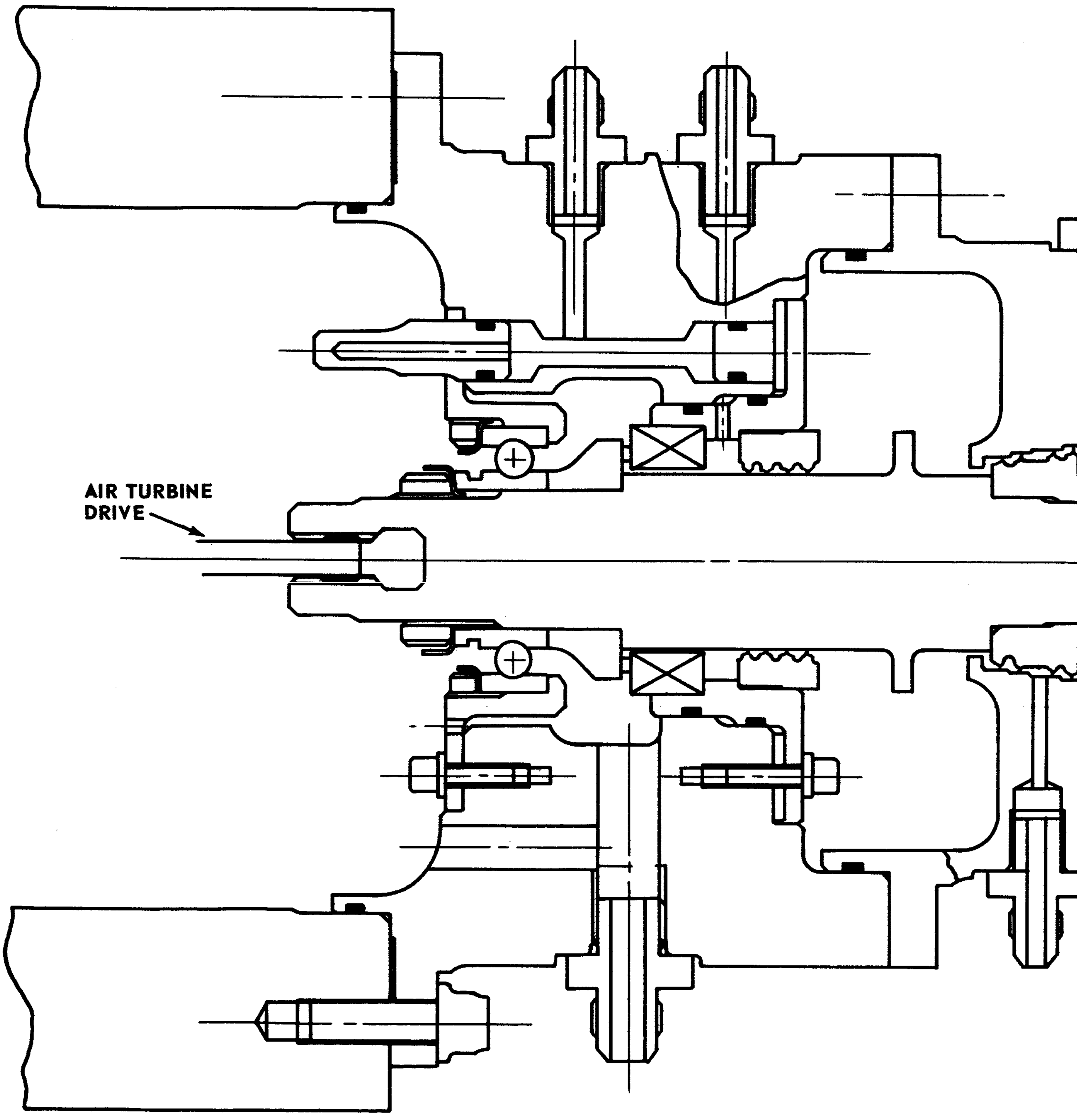
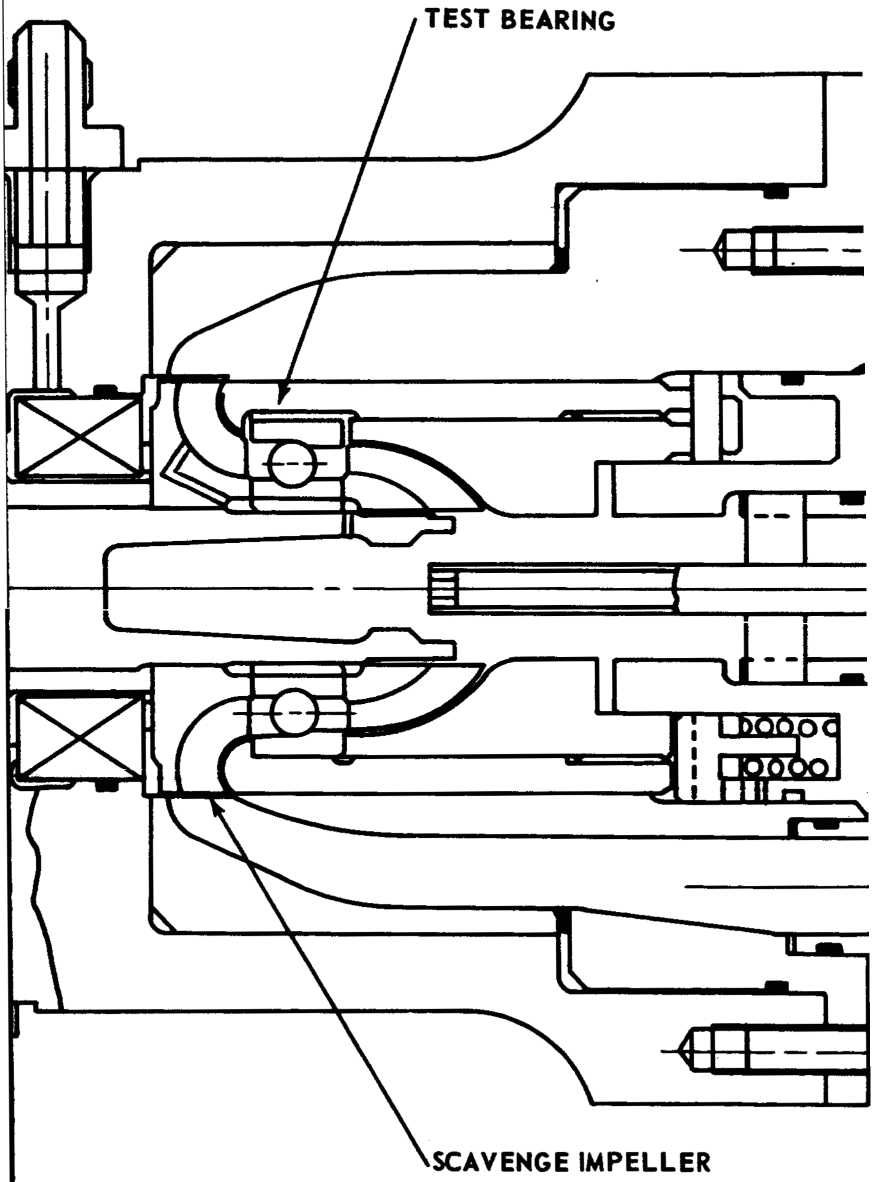


Figure 44 Turbine-Compressor Bearing-Scavenge Rig



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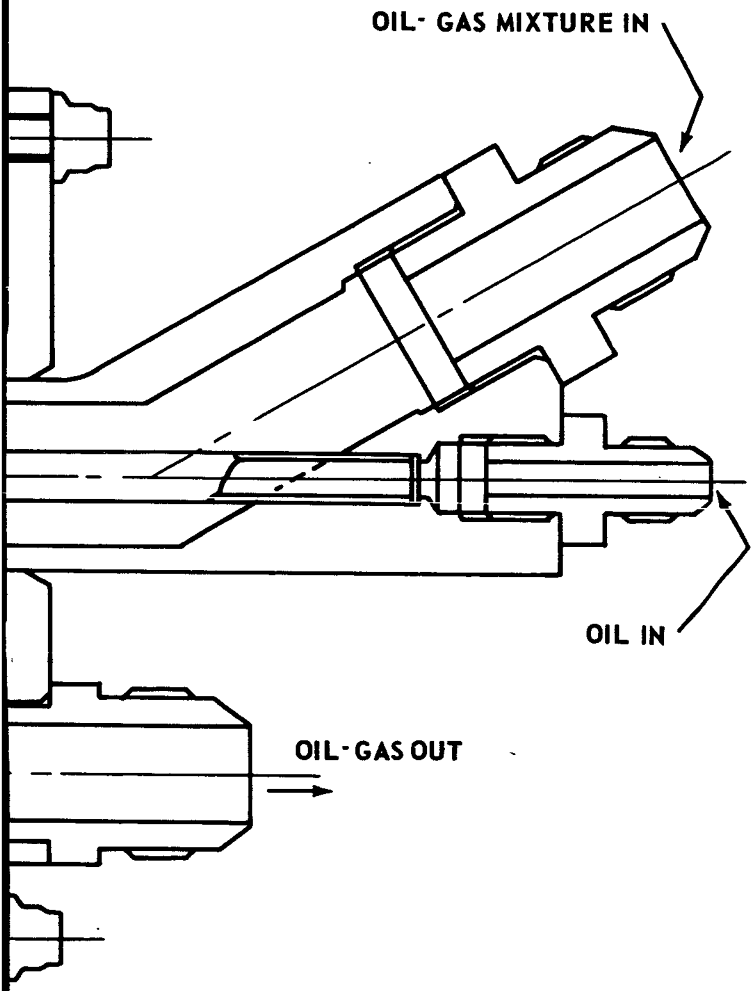


3

Design work is nearing completion on this seal rig with fabrication scheduled to follow as soon as drawings are available.

The purpose of the configuration shown in Figure 44 is to evaluate the performance of the turbine-compressor bearing, the scavenge pump, and the combination of bearing, seal, and scavenge pump. The design of this rig utilizes the same drive turbine and test adapter as the seal rig. The drive shaft design is similar to that of the seal rig but due to the inboard location of the seal the first rigid-body critical speed occurs at a higher speed, about 73,000 rpm. In the bearing test configuration, bearing cooling and lubrication will be investigated at various speed levels from 0 to 60,000 rpm, with most of the testing at 50,000 rpm. Bearing loads, shaft speed, oil flow, oil temperature rise and bearing temperatures will be determined. Power consumption will be estimated from the test data. Visual and dimensional nondestructive inspection of test parts will be made to determine the adequacy of the basic bearing design.

Since scavenging oil from bearing compartments with a minimum amount of power consumption is of major concern, several scavenge tests will be made to investigate the problems of scavenging the oil-argon mixture from the turbine-compressor bearing cavities. Two types of scavenge pumps are being designed; a conventional type centrifugal impeller, and the annular jet pump discussed on Page 21. In both cases the pumps will be pumping a mixture of gas and oil. Tests will be run at various shaft speeds including 12,000 and 50,000 rpm with the shaft in both horizontal and vertical attitudes. Oil and argon flow into and out of the bearing compartment will be monitored and varied at each speed. Photographic observation will be made of the exit mixture flow.

The test region of the bearing seal scavenge rig is versatile and will accept the complete turbine-compressor bearing cartridge including the squeeze film support and the axial springs.

VII. SEPARATOR-PUMP RIG

The oil-gas separator and oil pump are two components, in addition to the turbine-compressor bearing, seal, and scavenge components, which have been selected for experimental investigation. Both components are mounted on the free end of the turboalternator and operate at 12,000 rpm.

A separator-pump test rig is being designed to evaluate pump performance, separator performance, and combined pump-separator performance. The general design configuration for the separator-pump test rig shown on Figure 45 is similar to that of the bearing, seal, and scavenge test rigs. This test rig will be driven by an air turbine unit similar to that of the other rig. The test components are overhung from an intermediate housing as in the other test rig. The rig configuration presented in Figure 45 combines a scoop pump and a separator section. First-stage separation will be accomplished at the scoop pump where the bearing compartment scavenge gas-oil mixture is introduced, filling the shaft reservoir with oil and bypassing the gas, oil vapor, and overflow oil to the separator. Oil is pumped from the shaft reservoir to the accumulator by the scoop pump. The argon and some of the oil flows into the separator where the oil and gas are separated by centrifugal forces. The separated oil is pumped along the sloping outer wall and discharged forward toward the bearing compartment, while the argon is discharged from the rear of the separator and is piped out of the compartment. The test rig design provides the versatility necessary to evaluate pump and separator designs individually as well as simultaneously. Evaluation of pump concepts will be accomplished by removing the separator section and installing the pump test section shown in Figure 46. This test section will permit various gas-oil mixtures to be introduced into the shaft reservoir to evaluate scoop pump flow and pressure performance. First-stage separation efficiency at the scoop pump will be evaluated by measuring the oil-to-gas ratio of the pump discharge. The test rig is designed to provide the space necessary to evaluate a rotating oil cup reservoir pump which would have the oil scoop mounted in the periphery of the test section housing.

Separator designs will be evaluated by replacing the shaft oil reservoir feed assembly in the configuration shown in Figure 45 with a shorter oil-gas mixture feed line which discharges to the separator inlet only. Separator concepts will be tested by varying the inlet oil-gas mixture and measuring the oil content of the gas discharged from the separator. Transparent sections will be incorporated in the oil-gas mixture inlet line and gas discharge line to permit photographic records of the flow patterns to be made.

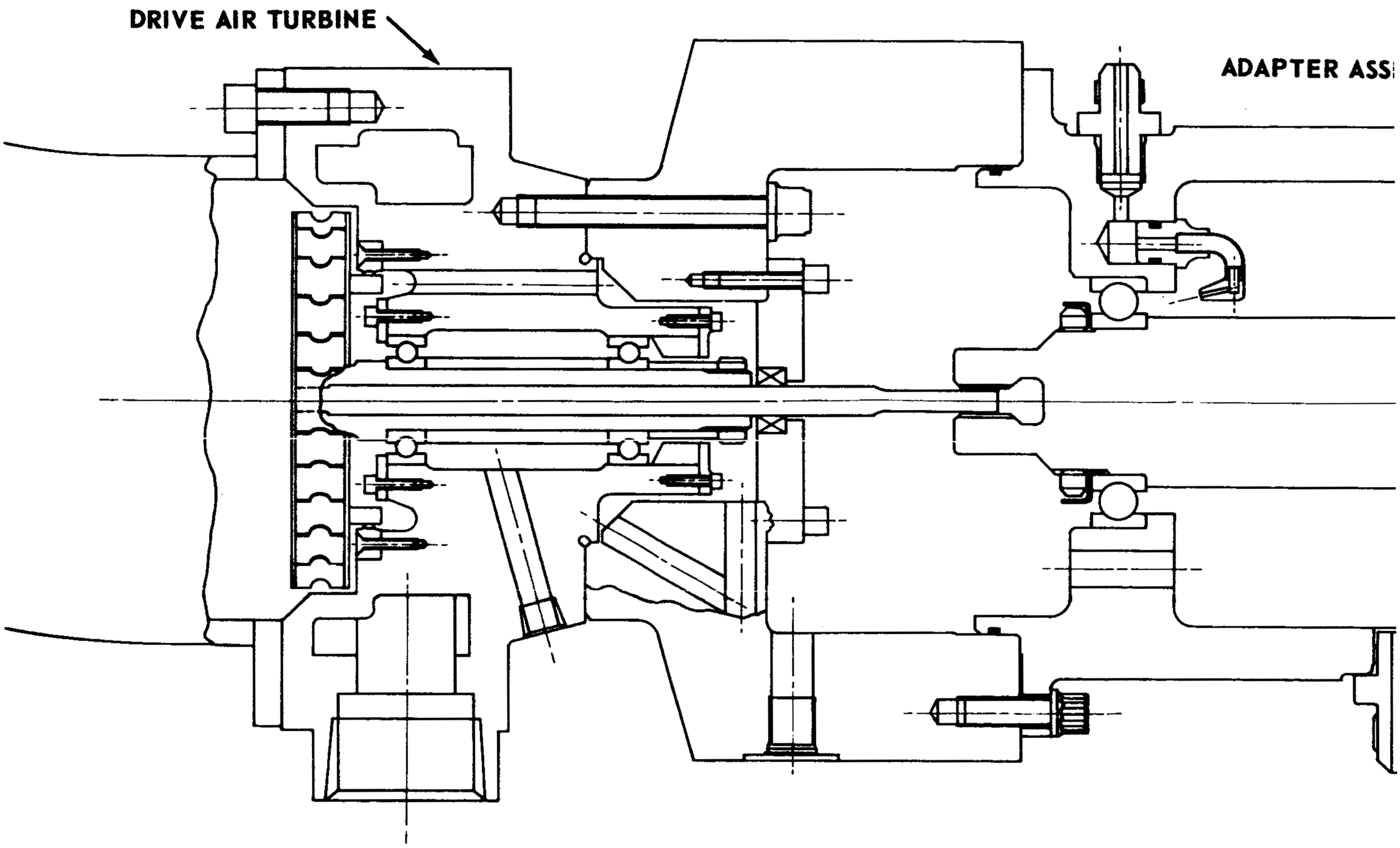
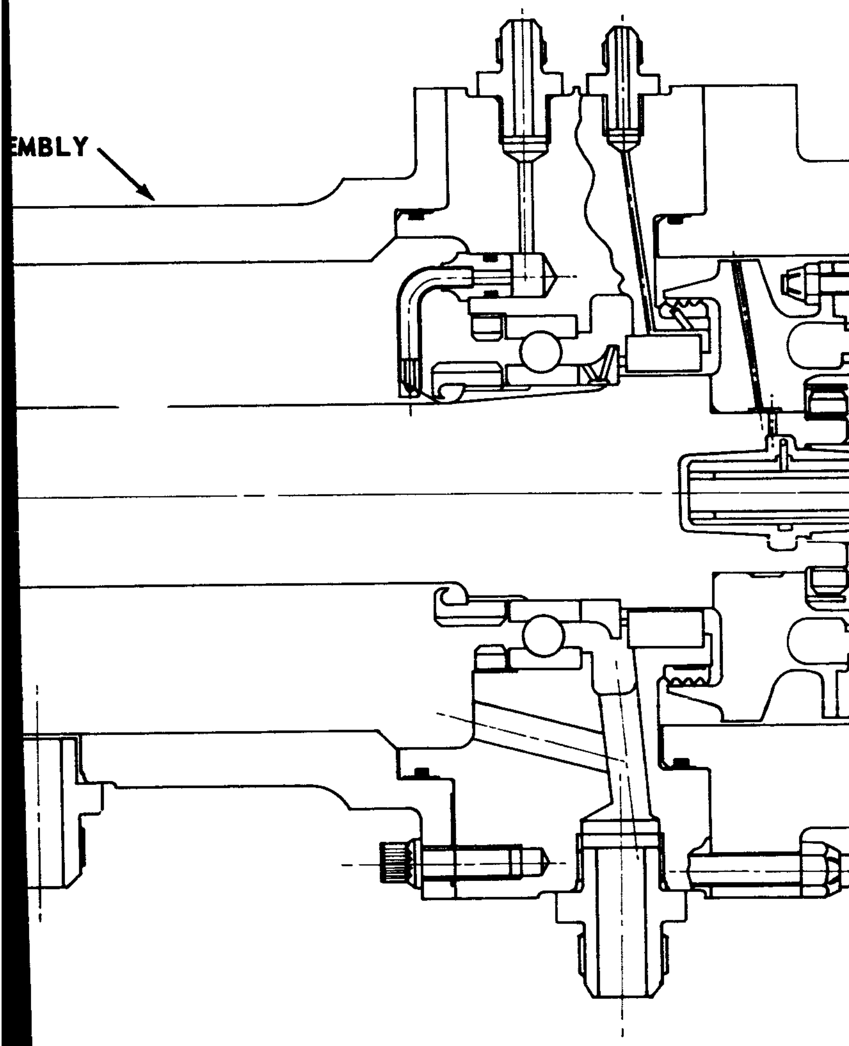


Figure 45 Turboalternator Pump-Separator Rig



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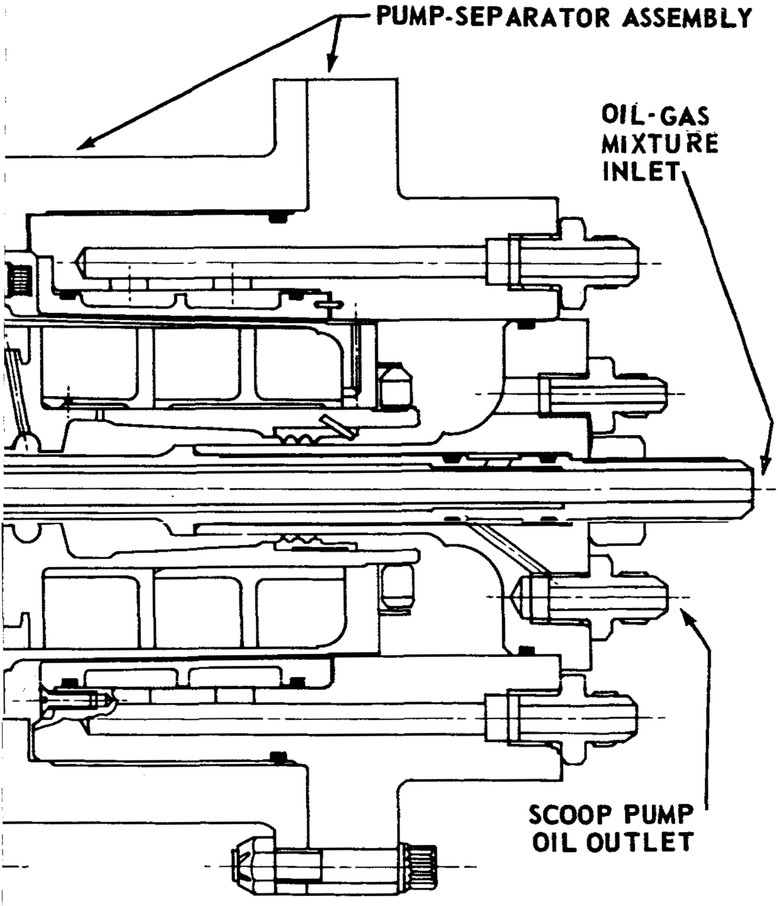
PUMP-SEPARATOR ASSEMBLY
OIL-GAS
MIXTURE
INLET
SCOOP PUMP
OIL OUTLET

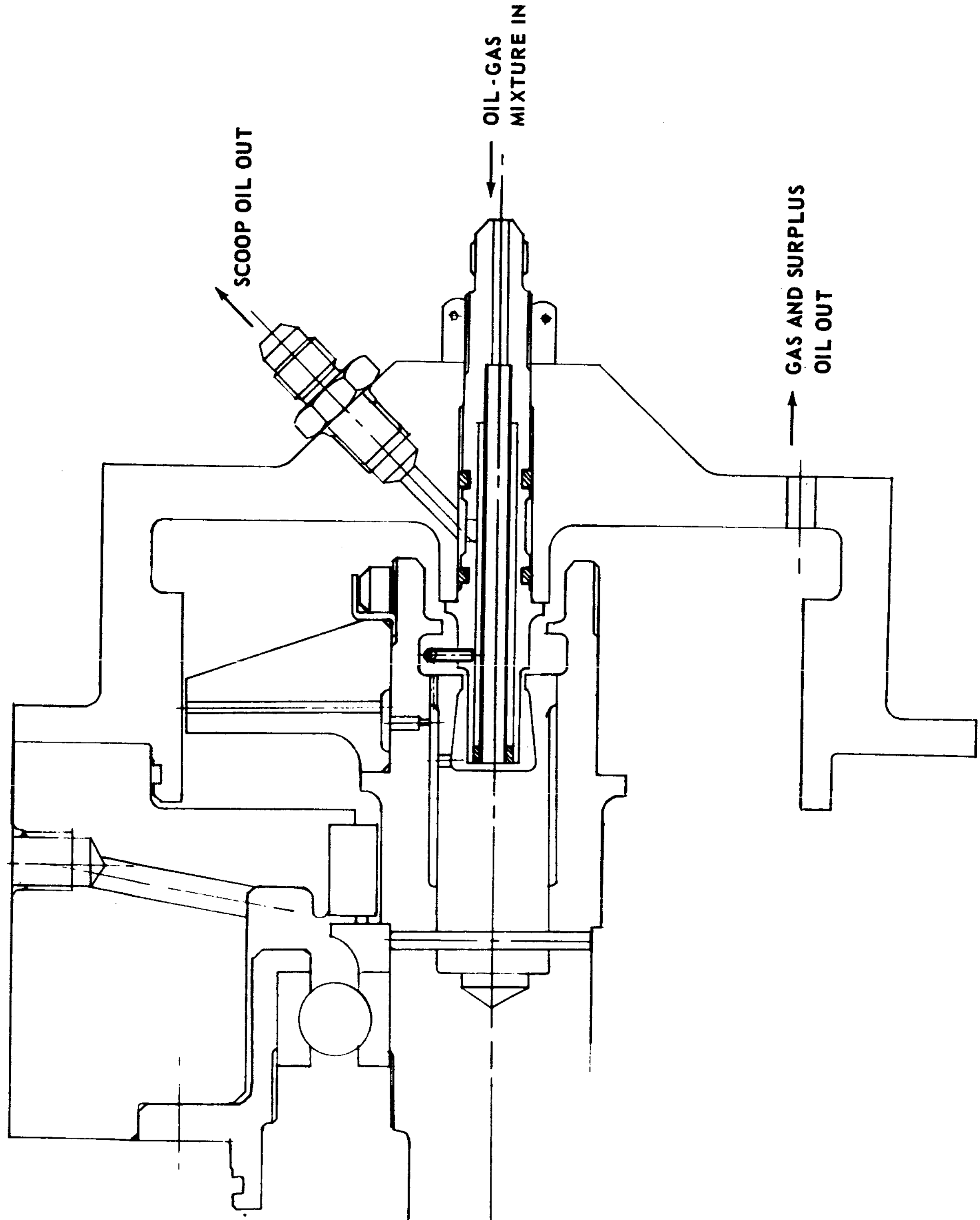

Figure 46 Pump-Separator Rig for Evaluation of Oil-Gas Separation and Pumping in Turboalternator Shaft

The rotor configuration shown in Figure 45 reflects the results of a design optimization to provide an adequate margin between the operating speed and the first critical speed. A bearing span of 5.14 inches was used in the initial design. A critical speed of 10,459 rpm was calculated. In order to elevate the first critical speed, larger bearings were used and the bearing span was increased to 8.00 inches. The first critical speed for this configuration is in excess of 17,000 rpm which is adequate for the test rig design speed of 12,000 rpm and specified overspeed of 14,400 rpm.

VIII. ADSORBER PROGRAM

The lubricant for the rolling-element bearings must be segregated from the high-purity main Brayton-cycle argon system. The design concept being pursued under this contract utilizes a combination of face seals and labyrinth seals to accomplish this separation. The labyrinth seals utilize a controlled flow of gas which is bled from the compressor. A portion of the bleed argon passes through the face seals into the bearing compartments and the remainder sweeps past the face seals to entrain any lubricant which migrates out of the bearing compartments through the face seals. This bleed argon must be purified before being returned to the main cycle. Oil in the form of droplets will be removed from the gas in a centrifugal separator which will also cool the gas to reduce the oil vapor solubility. Final purification of the argon will be accomplished by passing the gas through an adsorber bed prior to returning it to the main cycle at the compressor inlet.

A survey of available technical information on absorption of larger-molecule organic compounds has resulted in primary emphasis being placed on the use of molecular sieve materials for removal of lubricant from argon in the Brayton-cycle rolling-contact bearing system. The molecular sieve materials are alkali metal aluminosilicates which have been conditioned by removal of the water of hydration. Unlike most other hydrated materials, the physical structure of the molecular sieve crystal does not break down or reform when the water of hydration is removed. This characteristic results in a conditioned or activated structure which contains a network of interconnected cavities and passages constituting approximately half the total volume of the crystal. Three types of molecular sieve are commercially available. These are Linde types 4A, 5A and 13X. All three have a cubic structure. The 4A and 5A types have a cubic cell edge dimension of 12.32 angstroms while the type 13X has a cubic cell edge dimension of 24.95 angstroms. The structural configuration of the 4A, 5A and 13X sieve materials is such that molecules having critical dimensions of up to 4, 5 and 13 angstroms respectively can be admitted into the structural cavities and adsorbed. The 4A and 5A materials have a void volume of 0.28 cubic centimeter per gram and the 13X material has a void volume of 0.35 cubic centimeter per gram. The molecular sieve materials are available in 1/8 inch and 1/16 inch diameter pellets and in powder form. An inert material which represents 20 per cent of the total pellet weight is used to bond the pellets.

As reported in the first report[1], an adsorber test rig was designed and constructed. The test rig consists of a vaporizer, a swirl separator, and an adsorber column. Argon is bubbled through heated oil in the vaporizer to produce a mixture containing argon, oil vapor, and entrained oil. The gas-oil mixture leaving the separator is cooled to 100°F, corresponding to system adsorber inlet design temperature. This cooling causes precipitation of finely-divided oil particles in the gas stream which remain suspended in the stream. The gas stream is passed through a swirl separator prior to entering the adsorber test section to remove entrained oil droplets. This separator serves to simulate the function of the centrigugal separator attached to the turboalternator. After leaving the separator, the gas stream which contains oil vapor and aerosol passes through the adsorber column. The configuration of the test rig is shown in Figure 47. The tubing connecting the vaporizer to the adsorber column as well as the adsorber column are wrapped with electrical resistance tape to provide the heat necessary for temperature control of these components.

Testing was initiated using highly-refined four-ring polyphenyl ether and Linde 4A molecular sieve to investigate the operating characteristics of the test rig to develop instrumentation techniques. The gas pressure and flow rate entering the vaporizer were varied over a range from 1-5 psig and 1-4 liters per minute respectively at a constant vaporizer oil temperature of 350°F. The vaporizer oil temperature was varied from 290°F to 350°F at constant inlet gas pressure and flow to determine the effect of these variables on the concentration of oil in the gas entering the adsorber. The concentration of oil in the gas stream was found to be approximately 100 ppm by weight and no significant change in concentration was detected over the range of variables investigated. Apparently, the concentration of oil in the gas stream was dependent upon an equilibrium condition which was established as the gas and oil vapor leaving the vaporizer were cooled to adsorber inlet temperature. This characteristic permits operation of the test rig at a constant oil flow rate into the adsorber with an adequate tolerance on pressure and temperature control at the vaporizer. It also permits the flow rate of oil into the adsorber to be varied in proportion to the gas flow rate.

Adsorption evaluation tests of Linde 4A and 13X molecular sieve materials were performed using the adsorber test rig. The sieve materials were in powder form having particle size corresponding to 60-80 mesh or 1/16 inch pellets. In each test the adsorber column was loaded with approximately 36 grams of the molecular sieve material. The test rig was operated with four-ring polyphenyl ether heated to 350°F in the vaporizer with an argon flow rate of 1 liter per minute.

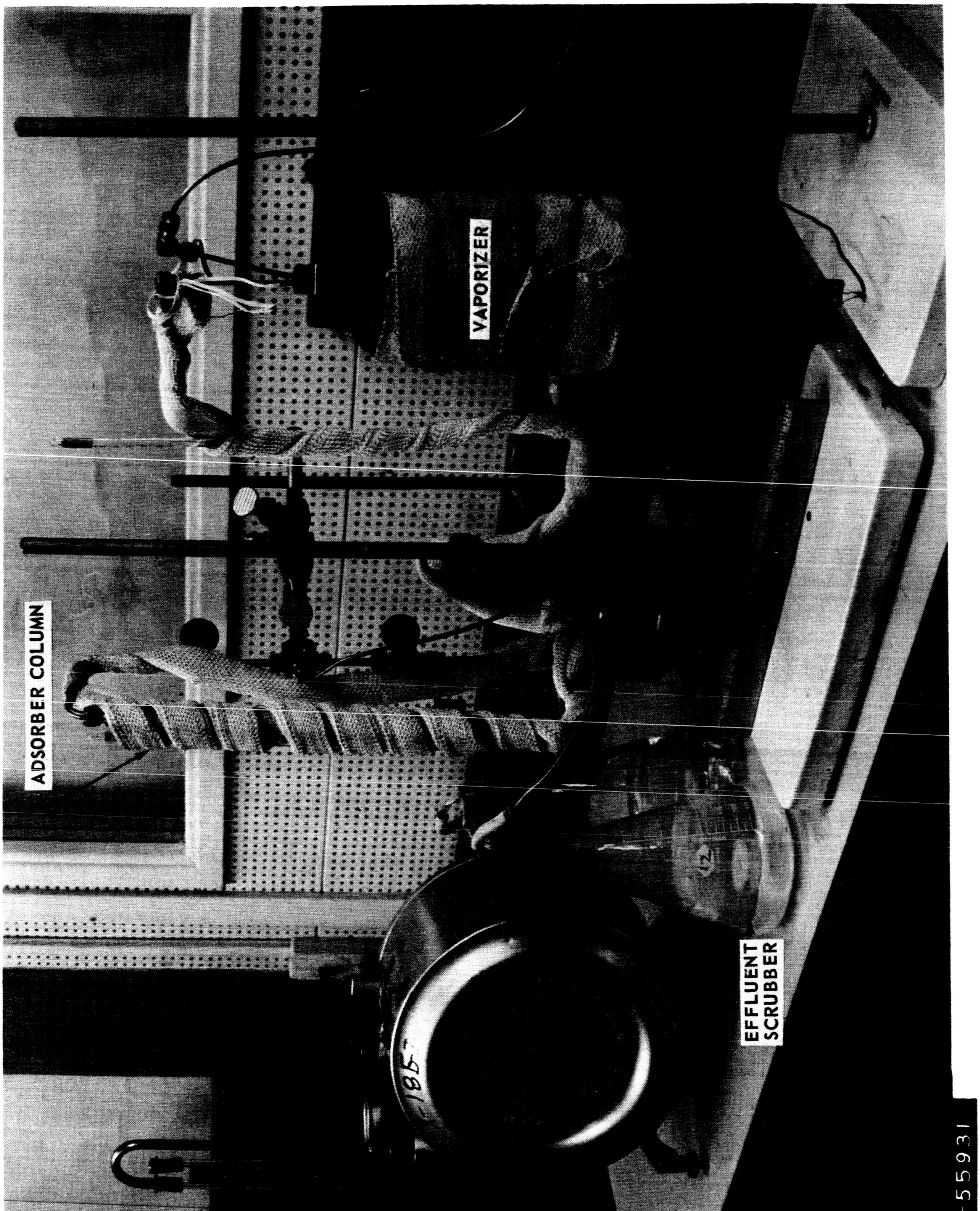

Figure 47 Modified Test Apparatus for Adsorbate Evaluation

The duration of each test was 100 hours. No oil was detected in the argon downstream of the adsorber column during the tests. At the end of the test period the contents of the adsorber column were divided into sections and the oil content of each section was extracted and measured. The results of the adsorber column oil content analysis for the first two tests were as follows:

100-Hour Test of Linde Type 4A Molecular Sieve Powder

Portion of Column	Grams of Oil Adsorbed	Micrograms of oil per gram of argon
first 1/4 (inlet)	0.746	61
second 1/4	0.521	42
third 1/4	0.025	2
remainder	0.003	0.3

100-Hour Test of Linde Type 13X Molecular Sieve Powder Conditioned at 300°F for 15 Hours

Portion of Column	Grams of Oil Adsorbed	Micrograms of oil per gram of argon
first 1/6	0.885	72
second 1/16	0.039	3
second 1/8	0.022	1.8
third	0.0012	0.1
remainder	less than 0.0008	less than 0.07

A post-test closeup view of the inlet section of an adsorber column is shown in Figure 48. In this test the adsorbent was molecular sieve having a particle size corresponding to 60-80 mesh. A band of oil-saturated adsorber is evident at the column inlet.

The test results of the second test would indicate that approximately 5 grams of 13X molecular sieve powder per gram of oil would remove essentially all of the polyphenyl ether entering the column. However, column pressure drop is inversely proportional to particle size and the pressure drop for the powder absorber is excessive. Therefore, to meet pressure drop requirements, the use of a pellet material may be more attractive.

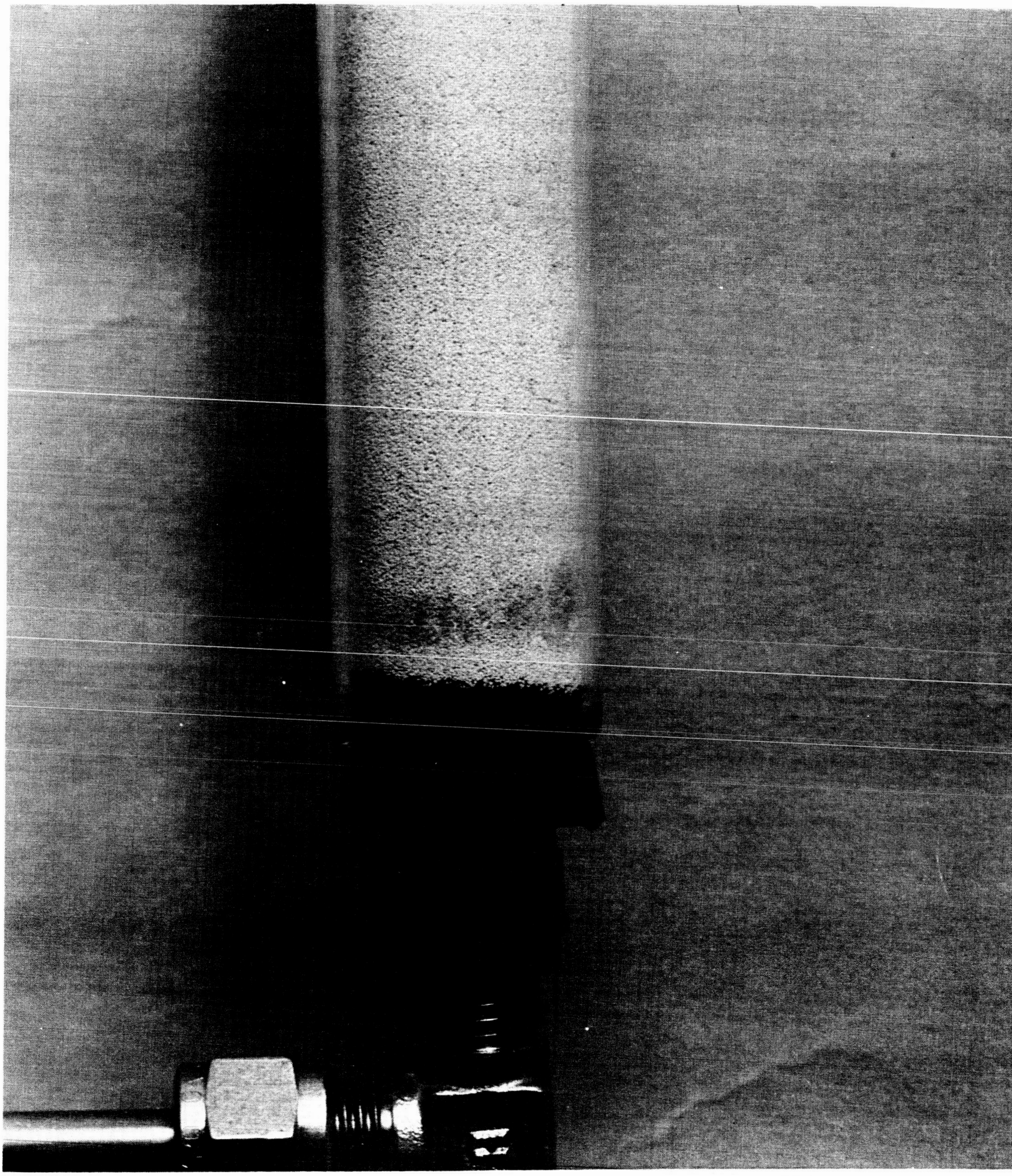

Figure 48 Inlet Section of Adsorber Column Filled with Adsorbate

A third 100-hour evaluation test was conducted using the adsorber test rig. In this test the adsorber column was loaded with approximately 40 grams of 1/16 inch diameter Linde 13X molecular sieve pellets. The test rig tubing and vaporizer were cleaned and the vaporizer was recharged with PWA-524 5-ring polyphenyl ether-based oil. The 100-hour test was completed. Analysis of the oil content of the adsorber material in the column is being conducted.

Particle size measurements on Linde Type 4A and 13X molecular sieve material in powdered form graded to 60-80 mesh size were completed. The results of these measurements are as follows:

Molecular Sieve Type	Size Range Microns
4A Pretest	131 - 188
4A Posttest	104 - 224
13X Pretest	83 - 166

IX. LUBRICANT SELECTION

In the previous report[1] the properties of three candidate lubricants were examined for this application. The three lubricants considered were 1) five-ring polyphenyl ether (PWA-524A), 2) four-ring polyphenyl ether (MCS-333), and 3) super-refined mineral oil (MLO-7277). During this report period one lubricant property was examined further, that is, the ability of the lubricant to provide long bearing life.

A direct comparison of the lubricant properties of the polyphenyl ethers and mineral oil is not possible based on the data available. Unfortunately, no test of a direct comparison of the oils has been reported. Some tests have been conducted under similar circumstances, but in these cases one oil contained additives and the other did not. The only experimental data of the performance of the oils with the bearing material selected, vacuum melt M-50, are reported in Reference 20. This paper presents the results of endurance tests of 30 M-50 bearings lubricated with PWA-524, 5-ring polyphenyl ether oil. The oil-in temperature was 500°F and the bearings were run at speeds between 1.25 and 1.5 million DN (diameter in millimeters x speed in rpm). Two different bearing sizes were tested. In the course of these tests none of the bearings failed, indicating a bearing life at least 14 times the AFBMA standard life. Unfortunately, comparable data with the other two candidate oils is not available.

Since the 5-ring polyphenyl ether (PWA-524A) oil has produced excellent results in vacuum melt M-50 bearing endurance tests, and since this oil has low vapor pressure minimizing the load on the adsorber, this oil is recommended for the Brayton-cycle application.

APPENDIX 1

References

APPENDIX 1

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